PUMPING AND HOISTING WORKS

PRESENTED BY

RISDON IRON WORKS

San Francisco, Cal.

1877.

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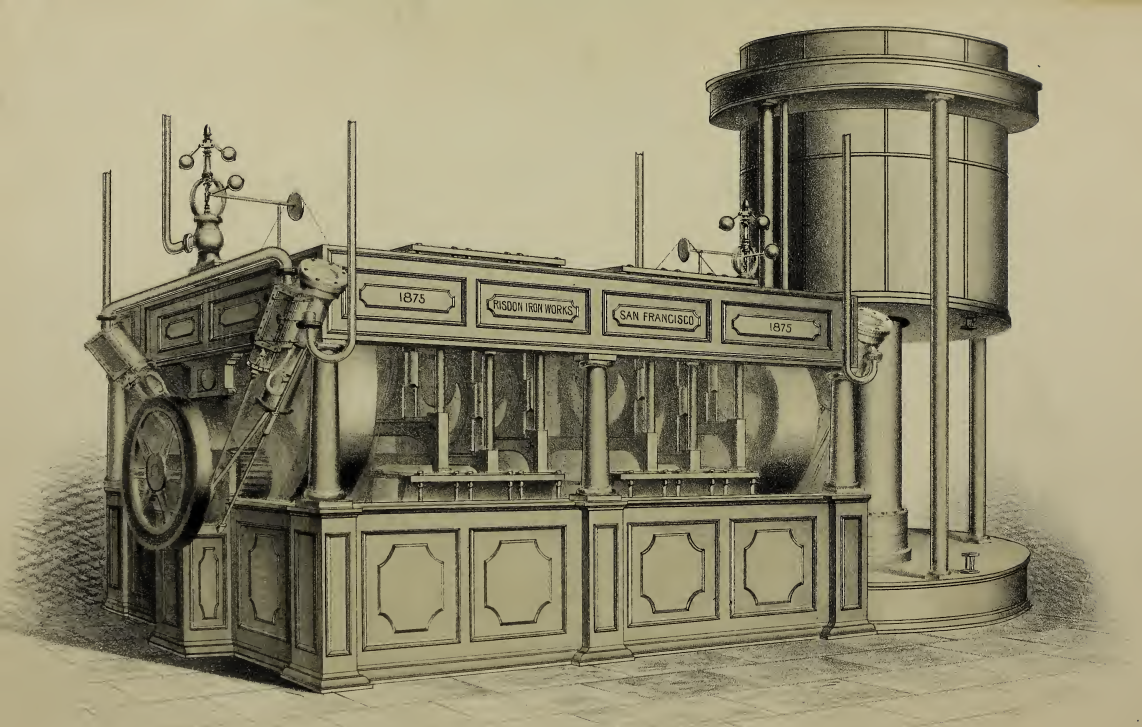

INTERIOR OF ACCUMULATOR HOUSE.
PALACE HOTEL, SAN FRANCISCO.

PUMPING AND HOISTING WORKS

FOR

GOLD AND SILVER MINES.

BY

JOSEPH MOORE, M.E.,

AND

GEORGE W. DICKIE, M.E.

PRESENTED TO.....

BY THE

RISDON IRON WORKS,

SAN FRANCISCO, CAL.

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A. L. BANCROFT & COMPANY, PRINTERS AND LITHOGRAPHERS.

1877.

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PREFACE.

The proprietors of the Risdon Iron Works, in presenting this work to their friends and patrons engaged in carrying out and superintending the class of work which it endeavors to illustrate, have been simply actuated by the desire to furnish them with a reference-book containing the information for which they have so often to apply to ourselves or others engaged in the construction of such works.

All the illustrations have been engraved on stone, in our own office, and directly from the drawings, under the personal supervision of Mr. Dickie, who, for many months, has given all his spare time to this work. The text of the work may be said to be the joint experience of Mr. Moore and Mr. Dickie, gathered, as they have labored together, in the same field of engineering.

This work not being in the shape of an advertising circular, we have endeavored to keep it free from any laudation of our own practice; nor do we expect it to be free from fault. Mining superintendents will, no doubt, find many things stated in this work to be the best that can be devised that they can improve upon. That part of the work which endeavors to point out what should be the future practice in this important branch of engineering, we know to be directly opposed to the expressed opinions of some leading engineers on the Comstock. We do not expect our opinions to decide what kind of machinery will be built to drain the great depths to which these mines may be wrought, but they may help a company to decide what is the best thing for them to do under certain circumstances.

This, the result of our labors, whatever value it may possess, we present to the mining engineers of California and Nevada; amongst whom we number many personal friends—men well qualified to judge of and form opinions upon the various subjects that we have endeavored to treat.

THE RISDON IRON WORKS.

W. H. Taylor,
President.

Joseph Moore,
Vice-President and General Superintendent.

PUMPING AND HOISTING WORKS

FOR

GOLD AND SILVER MINES.

Part First.

PUMPING ENGINES AND PUMPS.

The various constructions of pumping engines in use for draining mines and other purposes, where it is necessary to raise water from a lower to a higher level, may be classed as single acting direct, double acting direct, double acting, geared compound direct, and compound geared; and these are again divided into beam engines direct, and beam engines geared; horizontal engines direct, and horizontal engines geared.

The single acting Cornish beam engine, a sectional elevation of which is given in Plate 1, first claims our attention, not because it is popular among us, for there is not a single example of the Cornish engine among all the pumping machinery at work on our mines, but because it is the father of pumping engines both as to age and reputation, and it has never failed to show a good record wherever the principle upon which it depends for success was understood and faithfully carried into practice. If the theory of the survival of the fittest holds good in the world of mechanics, the Cornish pumping engine may take the lead of all others, for it has outlived many competitors for its position at the head of the mine shaft. It has been developed from the atmospheric engine, which was the earliest form of pumping engine employed in mining operations.

Why the Cornish engine has not obtained favor on this coast, may be accounted for in several ways.

First. All our mines are more or less prospecting enterprises, the ultimate depth to be attained and the water to be handled both unknown quantities; and in designing a Cornish engine both these quantities require to be known.

Second. The Cornish engine being single acting, the great size of cylinder required for a given amount of work renders it a very costly machine.

Third. The cost of mason work in this country, and the heavy stone work required for this form of engine. When these objections are cleared away the Cornish engine may yet have a trial with us.

One of the names that deserves to be remembered among the many Cornish celebrities for the part he took in the development of the Cornish engine, is that of Richard Trevithick.

About the year 1806 Trevithick attempted to introduce his simple non-condensing engine, both for pumping and hoisting, in place of Watt and Boulton's condensing engines. He considered that there being no need of condensing water would be a reason for their introduction; but that reason, where it was the only one, was not sufficient to warrant their adoption. Trevithick received several orders, but the "puffer," as it was called, did not become a general favorite.

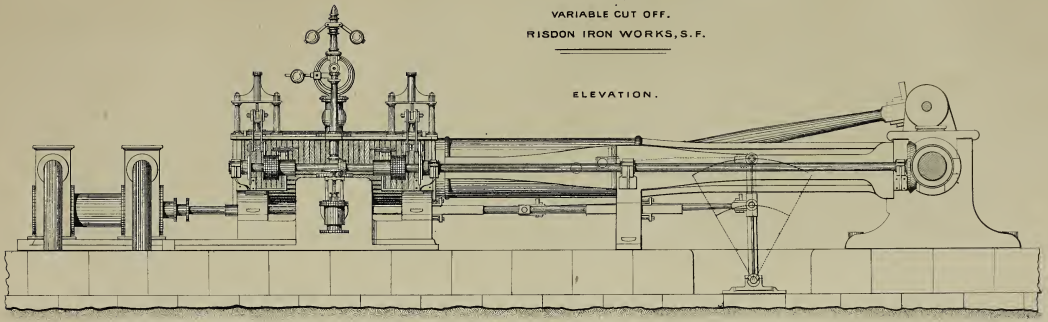
About this time McLean began to publish monthly reports of the duty performed by the pumping engines, and this stimulated a spirit of rivalry among the Cornish engineers and manufacturers, who, now left to their own resources, designed and manufactured the engines required in the country.

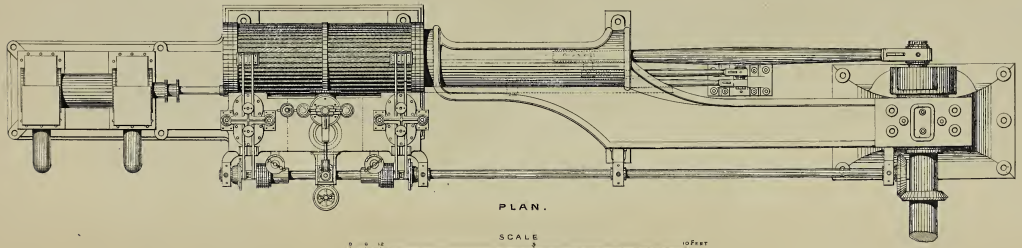
Trevithick soon took the lead, proposing the use of higher boiler pressures; the adoption of which, together with high degrees of expansion, has contributed more than anything else to the great increase of duty which marked the next few years in the history of the Cornish engine. The Watt and Boulton boilers were worked with steam only a little above the atmospheric pressure, as appears from the book of directions for working their engines, from which we cull the following: "When the engine is underloaded, it ought to be wrought with steam able to support one inch of mercury; and when fully loaded, it ought not to exceed two inches. In 1806 Trevithick proposed to Gilbert to use steam of 25 lbs. above the atmosphere, and to shut off the steam early in the stroke; so that at the end of the stroke the pressure should not exceed 4 lbs. absolute. Gilbert, in his reply, gave a theoretical analysis of the subject.

Trevithick, after thanking him for the calculations, remarked that he did not think that the pressures would hold good, because, on expanding the steam would get colder, and thereby lose a part of the expansive force.

HORIZONTAL CONDENSING ENGINE.
WITH
VARIABLE CUT OFF.
RISDON IRON WORKS, S.F.

ELEVATION.





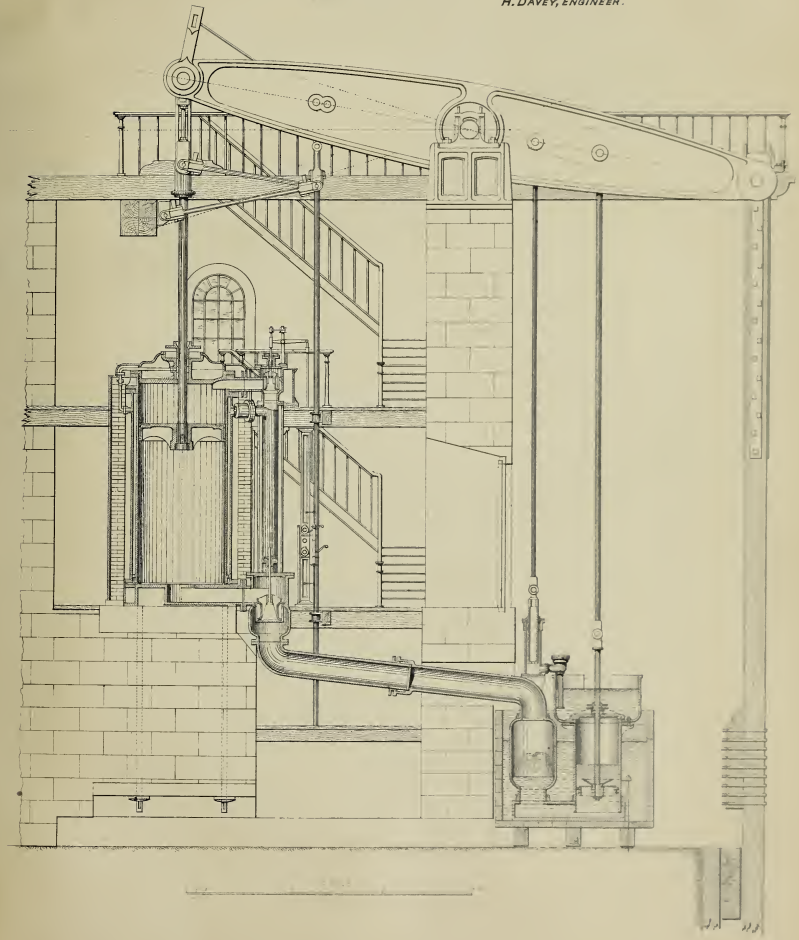
PLAN.

SCALE
0 6 12 10 Feet

CORNISH PUMPING ENGINE.

DIAMETER OF CYLINDER 60 INS. LENGTH OF STROKE 10 FT.

H. DAVEY, ENGINEER.



At that time nothing was known of the cooling effect, which takes place with the expansion of steam; but, instinctively it appears, Trevithick anticipated it. In fact, he proposed an engine and boiler almost exactly like those at present in use; and, in support of his proposition for the use of high-pressure steam, reasoned that the great mass of matter to be put in motion at the commencement of the stroke would, from its inertia (and momentum when in motion), act the part of a fly-wheel in regulating the motion of the engine. The Cornish adventurers were slow to aid Trevithick in his schemes to improve the duty of their engines; and it was not until 1812 that he began to be successful. In the spring of that year he erected a small engine at Wheal Prosper Mine. This was the first engine which worked as Cornish engines are worked now, and with the same type of boiler now in use. The cylinder was 24 inches diameter, and 6 feet stroke. The condenser was near the bottom of the cylinder, and had an air pump provided with a 12 inch diameter plunger, and 3 feet stroke, worked by a small beam placed above the cylinder. The main beam was placed beneath the cylinder.

The load on the engine consisted of three lifts: one of 120 feet 6 inch plunger pole; one of 120 feet 10 inch plunger pole; one bucket lift 60 feet, 12 inches diameter. The total load on the piston was 20 lbs. per square inch. The boiler was 24 feet long and about 6 feet in diameter, with fire tube. The construction and dimensions were, as near as can be, similar to those now in use. The steam pressure in the boiler was 40 lbs. above the atmosphere; there was no throttle valve between the boiler and engine, so that the steam was admitted with almost the full boiler pressure on the piston. This engine, with its usual load, worked more expansively than any of more modern times, with one or two exceptions, the cut-off being one-ninth of the stroke. Taylor's engine, at the United Mines, worked for years with a cut-off of one-tenth the stroke. We can find no record of the duty performed by Trevithick's first engine. The great importance of high pressures and high degrees of expansion began to be recognized with the introduction of Trevithick's—now called the Cornish—boiler, and many attempts were made to improve upon the single cylinder engine. Arthur Woolf took out a patent in 1810 for improvements in high pressure expansive working engines. Woolf became engineer to several of the mines, and in 1814 he erected the first of his engines at Wheal Abraham Mine. According to the monthly reports, the average duty of this engine for the first four months was 32.6 millions. A defect was discovered in one of the nozzle castings, which being removed, the duty increased in the following year to

52.3 millions. In the following year he erected a 50-inch cylinder engine at Wheal Bar Mine, which did a duty of 50 millions. The success of Woolf's engines spread his fame for several years, but an evil day awaited him. He was requested to erect a large double cylinder engine at Wheal Alfred Mine, in 1824. The diameter of the large cylinder was 70 inches, and the stroke 10 feet. This engine would no doubt have been as successful as the former ones had not Woolf persisted in putting in his cast-iron boilers, which proved a decided failure, and in consequence the double cylinder, which is playing such an important part in the economy of fuel in our day, fell into disuse. The introduction of the Cornish boiler by Trevithick contributed more than anything else to the great increase of duty that has been effected. It appears that the engineers of those days, like those of to-day, in the midst of all their anxiety to improve the engine, paid but little attention to the boiler, which, after all, is of equal if not greater importance. The duty returns of the Cornish engines would have been more valuable, and doubtless would have saved much misapprehension, if they had been so constructed as to have given the efficiency of the boiler as well as that of the engine. In the old boilers of 1800, the grate bars were often placed 8 and sometimes 10 feet below the bottom of the boiler. A boiler at Dolcoath Mine was 24 feet in diameter and 24 feet high, and the furnace held 30 tons of coal at a time. With such a boiler, who could expect a high duty? Since Trevithick's day, the Cornish engine has been improved in matters of detail, but no radical change has taken place. Captain Samuel Grose took the lead in the march of improvement about the year 1825. He erected an engine at Wheal Hope, and there carried out some experiments with a view to prevent loss of heat by radiation by clothing the cylinders, nozzles, and steam-pipes with non-conducting materials. He also considerably increased the pressure of steam and obtained better results. In the year 1827, he erected at Wheal Towan an 80-inch cylinder engine, in which he embodied his improvements. Great care was taken in the construction of this engine, and it did an extraordinary duty of 87 millions in the month of April, 1828: the average duty for the whole year was 77.3 millions. This brings us to the period of greatest efficiency in the Cornish engine, and we need not trace its history further. For reasons to be hereafter discussed, the duty of Cornish engines has fallen off to a considerable extent.

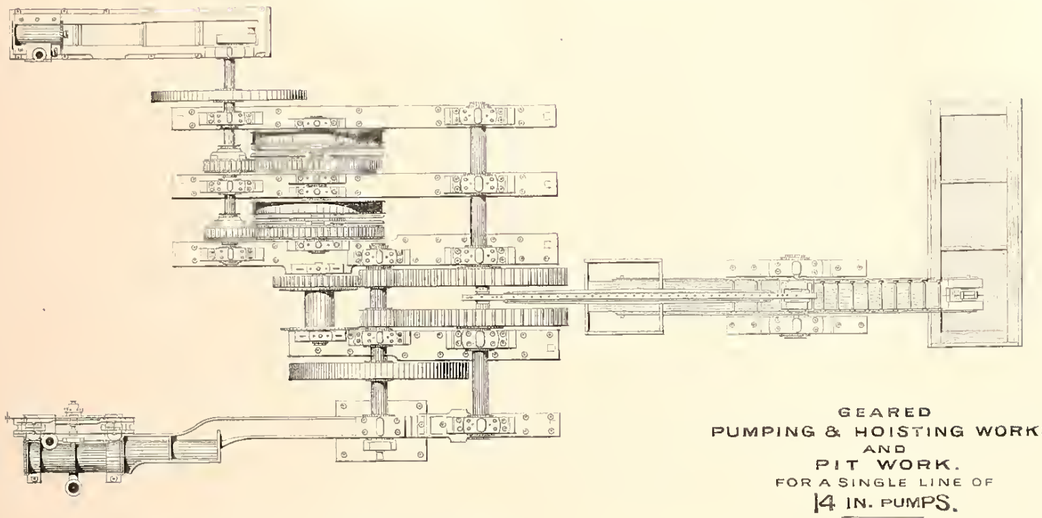
TABLE OF OBSERVATIONS UPON TEN CORNISH ENGINES IN 1838.

		Consolidated Mines.						United Mines.			
		Taylor's Engine.	Davey's Engine.	Job's Engine.	Woolf's Engine.	Bawden's Engine.	Pearce's Engine.	Curdnow's Engine.	Elden's Engine.	Loam's Engine.	Hocking's Engine.
Dimensions of Engine.	Diameter of Cylinder, in inches	85	80	65	90	90	65	90	30	85	85
	Stroke of Piston, in feet	10	$11\frac{1}{4}$	9	10	10	9	9	9	10	10
	Diameter of Steam Valve, in inches	12	13	9	8	8	7	10	5	10	12
	Diameter of Equilibrium Valve, in inches	16	18	12	16	16	12	13	7	16	16
	Diameter of Exhaust Valve, in inches	20	24	14	19	19	14	15	10	19	19
Dimensions of Boilers.	Number of Boilers	4	3	2	4	3	3	3	1	3	3
	Length of Boilers, in feet	3.36 1.40	37	31 32	35	36	36	36	36	1.32 2.38	44
	Diameter of Boilers, in feet	$6\frac{1}{2}$	7	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$
	Diameter of furnace flues, in feet	$3.3\frac{3}{4}$ 1.4	$4\frac{1}{2}$	$3\frac{3}{4}$	$3\frac{3}{4}$	$3\frac{3}{4}$	$3\frac{3}{4}$	$3\frac{3}{4}$	4	4	4
	Length of grate bars, in feet	4	4	4	4	4	4	4	4	4	4
	Total area of fire grate, square feet	63	52	30	60	45	45	48	16	32	48
	Cube inches in Cylinder, per foot of fire grate	10808	13145	11944	12722	16963	7963	14312	4772	21280	14186
	Total heating surface, in square feet	3781	3151	1598	3481	2694	2694	2694	941	2958	3451
	Cube inches in Cylinder, per foot of heating surface	180	217	224	219	283	133	255	81	230	197
	Water space, in cube feet	2467	2025	1033	2140	1650	1650	1650	579	1706	2085
	Steam space, in cube feet	735	580	315	608	468	468	468	178	528	645
Temperatures. *Fahrenheit.*	Open Air	57°	57°	—	57°	57°	51°	55°	56°	51°	55°
	Engine House	—	—	—	—	—	67°	—	65°	63°	66°
	Ashes over Boiler	80°	80°	81°	98°	98°	97°	88°	99°	79°	82°
	Cylinder Cover	111°	90°	102°	126°	96°	100°	94°	98°	90°	84°
	Middle of Cylinder Clothing	W77°	W76°	B95°	B102°	B140°	—	B79°	W60°	W67°	W68°
	Clothing of Steam pipe	79°	—	—	130°	95°	—	140°	97°	80°	82°
	Condensing Water	64°	84	58°	115°	110°	57°	60°	61°	65°	60°
	Hot Well	98°	100°	91°	140°	140°	97°	104°	94°	102°	96°
Height of Condenser Barometer, in inches		$27\frac{1}{4}$	$27\frac{1}{2}$	—	$25\frac{1}{2}$	—	—	—	—	—	$27\frac{1}{2}$
Number of Plunger Pumps		9	12	2	7	8	9	8	1	5	5
Number of Bucket Pumps		2	2	—	1	2	2	2	—	4	3
Proportion of stroke where steam is cut off		$\frac{1}{5}$	$\frac{1}{5}$	—	$\frac{1}{3}$	$\frac{1}{3}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{3}{5}$	$\frac{2}{11}$	—
Strokes per minute		$8\frac{1}{2}$	$7\frac{1}{2}$	—	9	9	10	$8\frac{1}{2}$	9	8	7

Having traced, in a brief way, the development of the Cornish Engine, which for mining purposes has been until very lately the standard style for all works of a permanent character in the Old World, let us now trace the steps by which we have arrived at our present position in pumping works on the Pacific Coast.

In the earlier days of mining in this country, neither great depths nor great quantities of water had to be dealt with. A simple water-wheel, generally a hurdy-gurdy, driven by a small jet from some high head, was the prime mover. A pump-bob, of the simplest construction, and a bucket-lift, all of wood, was the pit-work. Many a rig like this may be seen running yet where placer diggings are being worked. In quartz mines water-power was not often available, and steam had to be applied. On the Comstock mines, some of these pumping engines were of considerable power. The best of the older pumping engines, that of the Savage, a fine beam-engine of two hundred-horse power, geared five to one, driving a line of 12-inch plunger lifts to a depth of 1800 feet, is at present being removed to give place to a compound differential direct acting engine. The old engine, which is still a fine specimen of engineering, is we understand to be erected to drive the machinery of a quartz-mill. As the mines increased in depth and larger quantities of water had to be raised, large engines and heavy gears had to be built to do the work. Several of these engines, built in 1874–75, may be taken as representing the best that we have done with gearing and high-pressure expansive non-condensing engines. The pumping engine and surface work shown in Plate 3 is taken from the drawings of those built by The Risdon Iron Works for the Silver Hill, Dayton, Caledonia, New York, and Ophir mines. These five engines were all built from the same design, and of the following dimensions: Diameter of cylinder, 24 inches; length of stroke, 60 inches; fitted with double-beat valves and variable cut-off. Our illustration shows the arrangement of gearing, which consists of two pinions on the engine shaft gearing into two spur-wheels, the wrist-pin for the connecting rod to the pump-bob being secured in hubs on the arms of the spur-wheels. Every care was taken to have the works of the most permanent character; the foundation and bed-plates are of the most massive description. If these engines were all fitted with condensers, and the number of strokes made by the pumps kept within the limits of safe working, it would be a hard matter to beat them in economy of fuel.

In draining great depths, the great weight of the pit-work, which is of so much importance in any direct acting system, proves the destruction of the geared system. In driving from a wrist-pin, secured in the arm of a great

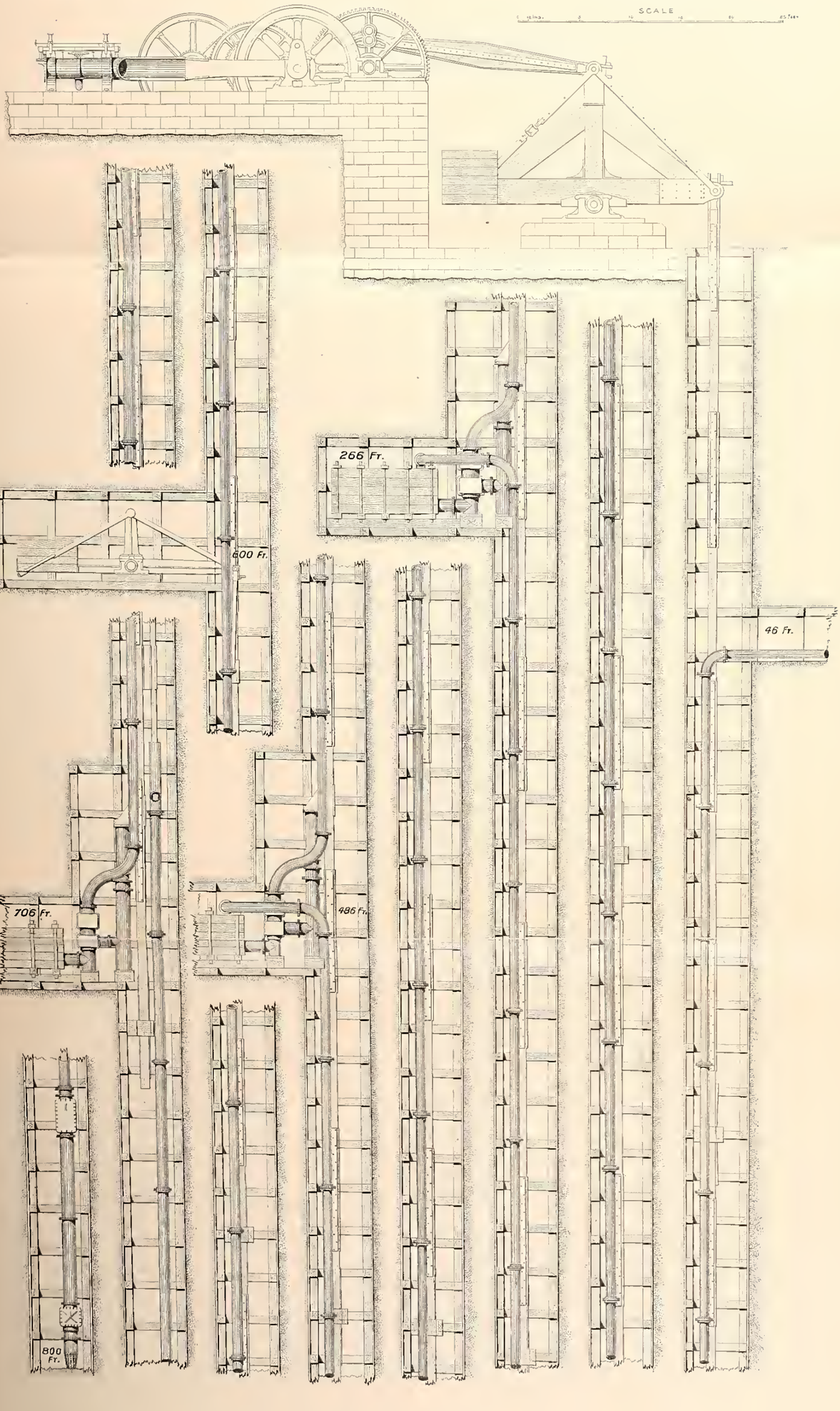


GEARED PUMPING & HOISTING WORKS, AND PIT WORK.

FOR A SINGLE LINE OF

14 IN. PUMPS.

SCALE



spur-wheel, the accelerating and retarding of the motion imparted to the great mass of pit-work must follow that of the wrist-pin, which it will only do at a certain rate of speed; this rate is determined by the weight of the mass set in motion, the speed or number of strokes, and the strength of parts. The consequence is that with geared pumping engines working at great depths with a heavy mass of pit-work to keep in motion the speed must be very slow indeed. Geared engines for great depths should therefore have the pumps made very large, and run very slow, otherwise the frequent breakages will soon condemn the whole system.

The pumping engine that at present claims our attention, because it is taking its place at the mouth of some of the deepest shafts in the Comstock, is that known by the name of Davey's Patent Differential Pumping Engine. In England this engine exists in two distinct types, viz., the single cylinder and the compound engine. On this coast it is being built in three ways.

First. The single cylinder, two of which have been built—one is at work at the Consolidated Virginia and California combination shaft, and the other at the Lady Washington shaft. In the cylinder of these engines the steam is expanded as far as the weight of the pit-work and the velocity of motion will admit, which in this case is not very much. After making the stroke the steam is exhausted into the atmosphere, as from an ordinary steam pump. We may here state that the engine on the C. and C. shaft has since been compounded.

Second. The compound condensing engine. Several examples of this class, both beam and horizontal, have been built; and we will again refer to some of them. In this class the steam is expanded to a greater extent than in the single cylinder; the use of two cylinders making this possible with a comparatively light load and small velocity. This class, admitting of being worked with high steam pressure and high degrees of expansion, is capable of realizing the greatest economy of fuel.

Third. The compound non-condensing engine, one or two examples of which have also been built. This is the same as the second style, without the condenser. Where there is the possibility of obtaining condensing water, this style should never be adopted. It may seem strange to suppose the want of condensing water in a pumping engine; but at several mines the water is discharged through a tunnel, sometimes as much as 150 feet from the surface, and it then may be a question whether the vacuum obtained will do much more than the extra work in lifting the water amounts to, and then in many cases the water from the mine is hot, sometimes reaching 125 degrees, and cooling ponds have to be resorted to in order to be able to

condense. All these difficulties, added to the light atmosphere at the altitude of most of our mines, makes it necessary for the engineer to consider what will be the value of a condenser to him. Hence, compound non-condensing engines have been built, as it were, under protest.

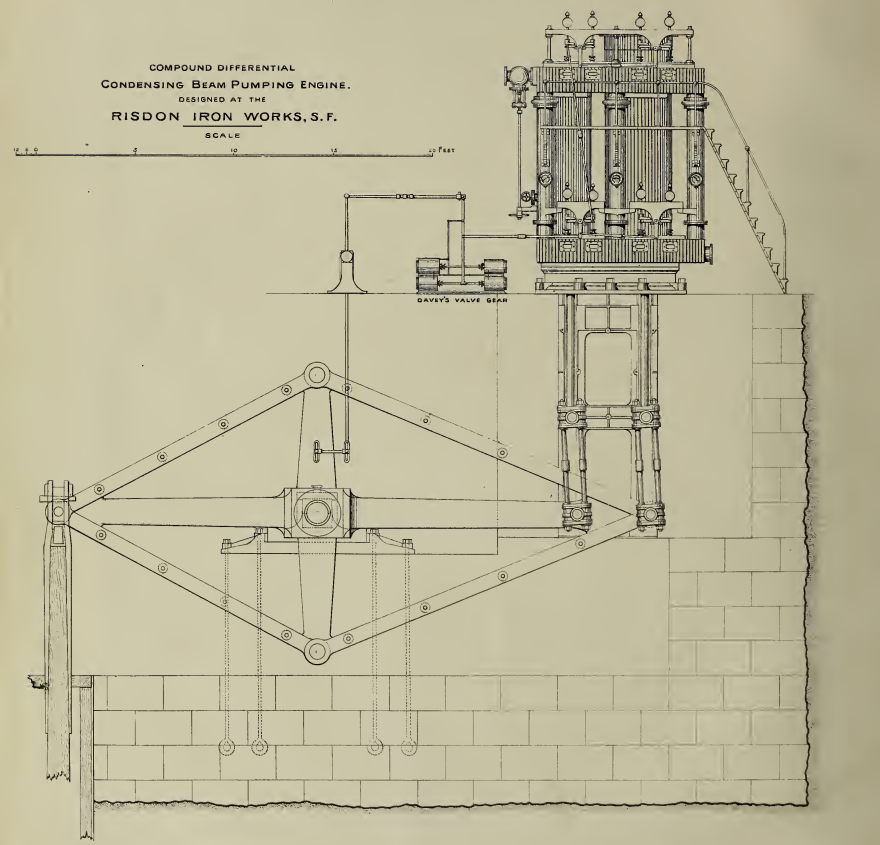
The chief peculiarity in the invention is the simple manner in which the engine is made perfectly safe in working, under all conditions of load, automatically and instantly varying its supply of steam with every minute increase or decrease of resistance; the distribution of steam being such that the pumping is performed without shock, even when the pressures suddenly and greatly vary.

Plates 8 and 9 give a front elevation, end elevation, and plan of a compound differential condensing beam engine, designed by us and built by the Risdon Iron Works for the Lady Bryan mine. Dimensions of this engine are: High pressure; cylinder 26 inches diameter, 8 feet stroke. Low pressure; cylinder 38 inches diameter, 10 feet 7½ inches stroke. The pumps have a stroke of 8 feet. The beam is placed below the cylinders, the centre and arms being of cast iron trussed with wrought iron straps, as shown. The high pressure cylinder is placed nearest the centre of the beam, and has the same stroke as the pumps. The longer stroke of the low pressure cylinder keeps down the diameter and preserves the symmetry of the engine. The valves are all of the double beat Cornish type—the top set for both cylinders being in one valve chest and the bottom set in another. These chests are joined by three columns—the one nearest the beam centre being the steam column, the middle one being the receiver and the outside one the exhaust. This arrangement gives the engine a fine appearance and keeps all the valve gear on one side, and in sight of the engine man. It is provided with a separate condenser, to which the hoisting engines will also be connected. This will, no doubt, be a very fine engine, and will work with the greatest degree of economy; but for an engine of these dimensions the horizontal type would be less expensive and equally effective. There have been other three-beam engines built—one by the Risdon and two by the Union Iron Works—the valves of which are operated by the differential gear. These engines are from designs by Mr. Patton, Consulting Engineer, Virginia City. They are of larger dimensions, and differ somewhat in arrangement from the one we have already described. The position of the cylinders is reversed—the low pressure cylinder being nearest the beam centre. Each valve has a separate chest for itself—four valve chests being on one side of the engine and four on the other—the levers and lifters for operating the valves being all arranged on the top of the cylinders. The

FRONT ELEVATION

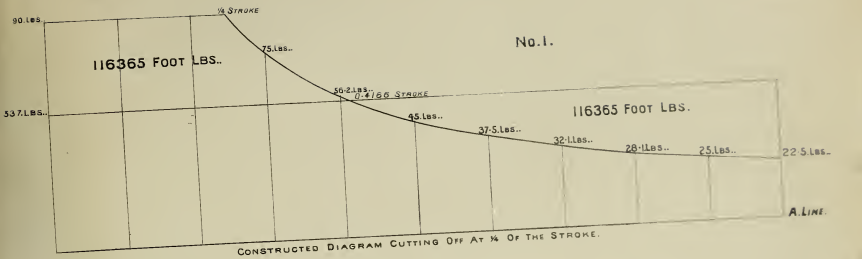
COMPOUND DIFFERENTIAL
CONDENSING BEAM PUMPING ENGINE.
DESIGNED AT THE
RISDON IRON WORKS, S.F.
SCALE

0 5 10 15 FEET



PLAN

END ELEVATION



Constructed Diagram Cutting Off At ¼ Of The Stroke.

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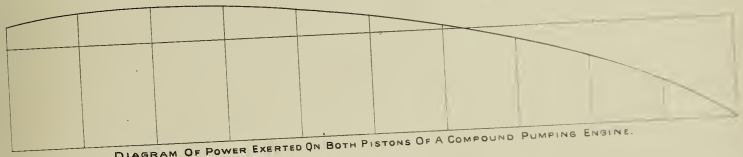
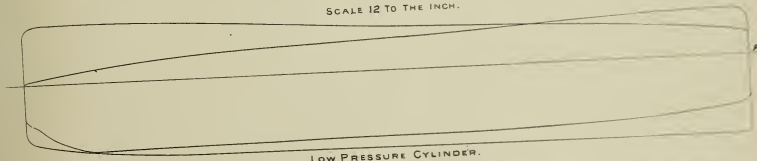


Diagram Of Power Exerted On Both Pistons Of A Compound Pumping Engine.

No. 3.

Scale 12 To The Inch.

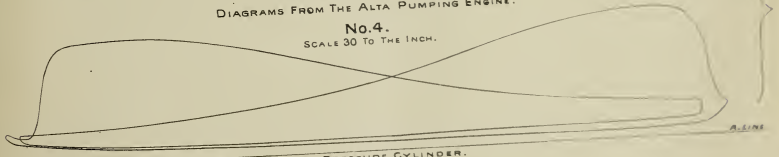


Low Pressure Cylinder.

Diagrams From The Alta Pumping Engine.

No. 4.

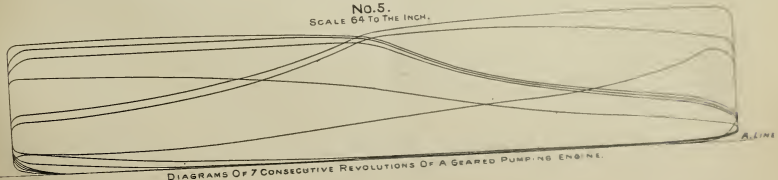
Scale 30 To The Inch.



High Pressure Cylinder.

No. 5.

Scale 64 To The Inch.



Diagrams Of 7 Consecutive Revolutions Of A Geared Pumping Engine.

beam is placed under the cylinders, and is double—of rather peculiar construction—having no compression members to the truss. Nothing has been spared to make these very fine pumping engines, and they will no doubt show a good record when they get to work. These engines are also fitted with separate condensers. (See Plate 15.)

We have already stated that the differential gear has been applied to single cylinder engines. These engines were designed by Mr. Patton in 1875. One is at work on the C. & C. shaft, and the other at the Lady Washington; they are duplicates of each other. Cylinders are 40 inches in diameter, and have a stroke of 8 feet, the stroke of the pumps being 7 feet. They have been at work about one year, but we have not been able to get any data from which to figure their economy. We hardly think it possible that they can be as economical as the best style of geared engine. In these engines the steam is not expanded to any extent; in fact, it is impossible to do so, because the piston speed must be limited. For the sake of illustration, let the engine make eight double strokes per minute, which would be a fair working speed, allowing time for pauses, the speed of the pit-work at that point of the stroke where the expansion curve besects the mean pressure line could not be more than 3 feet per second. We have not been able to procure any indicator diagram from these engines that show them working with any degree of expansion, and have, therefore, constructed one for them cutting off at one-fourth of the stroke. (See diagram No. 1, Plate 5.) Pressure in the cylinder at point of cut-off 90lbs. would require about 110lbs. pressure in the boiler. The mean pressure would equal 53·7 lbs. area of piston = 1256·64 × 53·7 = 67481·5 lbs. pressure stroke of 8 feet; stroke of pumps 7 feet = $\frac{67481{\cdot}5 \times 8}{7}$ = 77121·7, less 25 per cent. for friction = 57841·3 × 2, engine being double acting and pumps single acting = 115682; the pumps in this case are a single line of 12-inch plunger lifts.

Area of plungers = $\frac{115682}{113}$ = 1023lbs. pressure per square inch, equal to a depth of 2380 feet. This, of course, is the theoretical duty of the engine, the actual work would fall somewhat short of this depth.

It will be noticed that the mean pressure line on the diagram besects the expansion curve at ·4166 of the stroke; that is, when the piston has traveled 40 inches, the steam pressure upon it will be just equal or balance the load. The pressure before this point has been reached has been greatly in excess of the load, so that up to this point the motion of the piston has been accelerated, and the excess of power has been stored up to meet the

deficiency that will occur in the latter part of the stroke. By the diagram it will be seen that this deficiency just equals the excess in the first part of the stroke. Now let it be supposed that the piston has attained a velocity equal to 3 feet per second in the pumps after it has traveled 40 inches; there will then be stored up in the pit-work at that point of the stroke 116365 foot lbs. of work. This may be represented either by velocity or weight; but the velocity being fixed at 3 feet per second, we must have sufficient weight to absorb this amount of power.

The energy of a mass is expressed by the equation $\frac{W. v^2}{2\ G}$, therefore the mass to be set in motion, or the weight of pit-work required for this degree of expansion would be $\frac{116365 \times 64}{9} = 827484$ lbs., or a little over 400 tons as the least weight that would be required to enable this engine to work with the moderate degree of expansion that has been here taken for illustration, so that little or no expansion can be attained in working these engines with their present weight of pit-work and rate of speed. As these engines are working smaller pumps than will likely be used on any of the mines in the future, let us suppose a single cylinder non-condensing double acting pumping engine for a heavier class of work, say one single cylinder horizontal double acting pumping engine, non-condensing, using steam of 80 lbs. pressure above the atmosphere, steam to be cut off at $\frac{1}{4}$ of the stroke when working at 3000 feet. Pit-work to consist of two sets of $12\frac{1}{2}$ inch plunger pumps, 9 feet stroke. Weight of water-column on each pump = 152·170 lbs. + 25 % for friction = 190212 lbs. load for each side of piston. The mean pressure for 80 lbs. + 12 for atmosphere at the altitude of Virginia City $= \frac{(1 + \text{Hyp. Log. } 4)\ 92}{4} = 54{\cdot}878 - 12 = 42{\cdot}878$ mean pressure. Area of piston would therefore be $\frac{190212}{42{\cdot}878} = 4436$ square inches $= 75\frac{1}{8}$ inches diameter. The expansion curve on a constructed diagram would besect the line of mean pressure at ·425 to the stroke; that is, the pressure of steam in the cylinder will balance the load on the piston after it has traveled 3·825 feet. During the latter part of the stroke, the pressure on the piston being much less than the load, the power represented by that portion of the diagram above the line of mean pressure is required to make up the deficiency.

This power or energy is stored up in the moving mass represented by the spear-rods, plungers, balance-bobs, and the water-column.

The average pressure of that portion of the diagram above the mean pressure line is 30·26 lbs. × 4436 ÷ 3·825 = 513443 foot lbs. stored up energy,

the energy of the mass $= \frac{W.v^2}{2G}$. Therefore the mass to be set in motion would be $\frac{513443 \times 64}{16} = 2053772$ lbs., or a little over one thousand tons, with a piston speed of 4 feet per second, which, in our opinion, is a greater speed than is quite safe for double-acting pumping engines.

The actual moving mass would be:

Pump plungers and attachments	70,400
Weight of rising column	304,340
Weight of spear rods and irons	235,600
Balance weights	153,830
Moving parts of engine	41,000
Total weight in motion	805,170

The weight of the pit work being 805,170 lbs., an additional 1,248,602 lbs. would be required as dead weight; one-half on the spear rods, the other half on balance bobs, in order to carry out this limited degree of expansion, whilst the strains on the pit work would be enormous. The mean thrust on the pump spears would be 190,212 lbs., and the maximum thrust would be 354,880 lbs. We need hardly add that it is impracticable to work a single cylinder double-acting pumping engine with any degree of expansion that will prove economical.

Of course, single cylinder Cornish engines have worked with a very early cut-off, as we have already seen. In the early days of the Cornish engine expansion was carried to its utmost practical limit; but the risk incurred by the necessary velocity of the steam stroke, and the numerous accidents that followed in consequence led to the adoption of a lower and safer degree of expansion.

Other things being equal, economy of fuel is the most important question connected with pumping machinery, and the leading principle of economy is expansion, the engine that will safely work with the greatest amount of expansion is the most economical. There are, however, certain conditions necessary to expansive working.

The resistance to be overcome in pumping is almost constant, and the force applied to overcome that resistance by the expansion varies. The diagram of force from the pumps would be a parallelogram, and the force of the steam is a parabola. It is evident that the mean of the two forces must coincide; but the extremes greatly vary. A means, then, is required, whereby the work due to the portion of the parabola, above the boundary of the parallelogram, may be stored up to be added to the portion of the parabola

below the boundary of the parallelogram. That function is performed, as we have already seen in the Cornish engine, by the inertia and momentum of the pit work, beam, column, and other moving parts; and in the rotative engine by that of the fly-wheel. The elements of this compensating force, *vis viva*, are weight and velocity, expressed by the equation $\frac{W.v^2}{2G}$. It is evident to the casual observer that such a condition of things must cause an enormous strain, above the resistance of the pumps, to be put on the engine at the commencement of the stroke when a high degree of expansion is employed; and, further, that the piston speed, where the boundary lines of of the two figures cross each other, must be very great.

It follows, then, as corollaries:

1. That to expand, to a great extent, in a single cylinder, a great strain must be put on the engine above that necessary to overcome the resistance of working the pumps.

2. That a great moving mass must be provided, and a great maximum piston speed employed to render expansive working in a high degree practicable in a single cylinder.

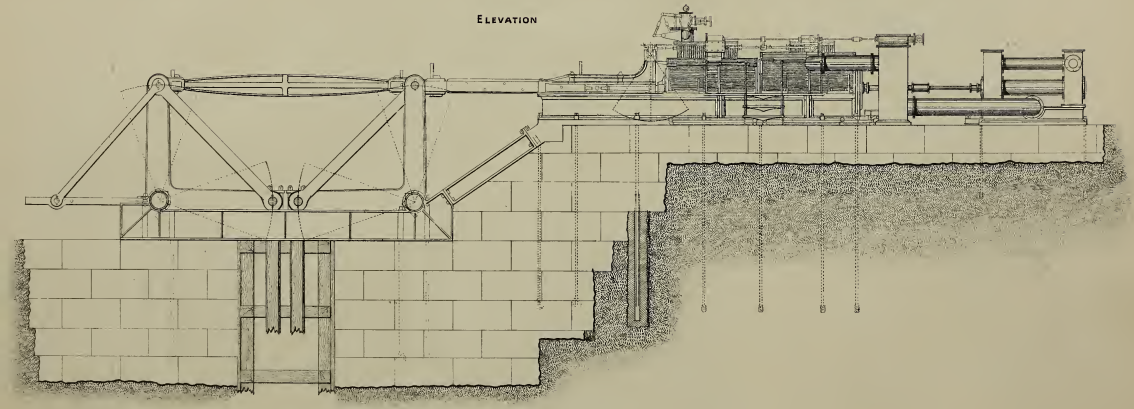
In practice, the first of these corollaries is illustrated by the many accidents which have taken place with Cornish engines working with a high degree of expansion; and, in consequence, expansion has of late years been very much reduced, causing a large falling off in duty.

The second is illustrated by the fact that it has not been found practicable to work Cornish engines expansively with a moderate piston speed, such as is necessary in lifting water. The only condition under which the Cornish engine has proved itself an economical machine is that of actuating plunger lifts where a great mass of matter has been lifted at a great speed during the steam stroke, and then caused to fall slowly during the equilibrium stroke in lifting the water in the pumps.

These two great difficulties, always present where pumping has to be done, are surmounted, to a certain extent, by the compound differential engine. The diagrams taken from these engines show that the variation in force between the commencement and end of the stroke is as $2\frac{1}{2}$ to 1. Whereas, in the single cylinder, with the same degree of expansion, it would be 6.3 to 1; that is to say, the variation in the two pressures is nearly three times as great in the single cylinder engine as that in the Compound Differential.

As this variation has to be compensated by weight and velocity, it is evident that with the smaller variation less weight and less speed are required.

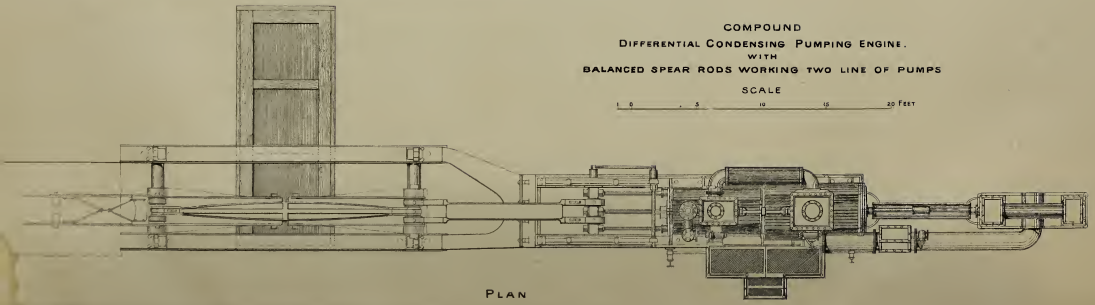
ELEVATION



COMPOUND
DIFFERENTIAL CONDENSING PUMPING ENGINE.
WITH
BALANCED SPEAR RODS WORKING TWO LINE OF PUMPS

SCALE

1 0 5 10 15 20 FEET



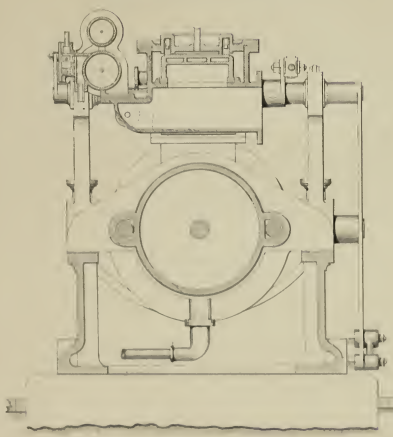
PLAN

DETAILS
OF
HORIZONTAL COMPOUND
DIFFERENTAL PUMPING ENGINE.

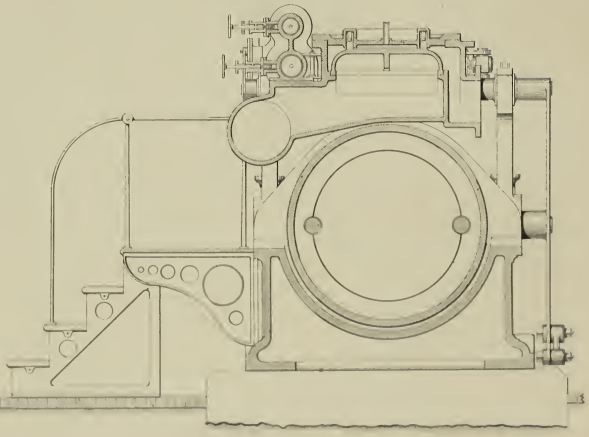
SCALE

0 1 2 3 4 5 6 Feet

CROSS SECTION
THROUGH
HIGH PRESSURE CYLINDER

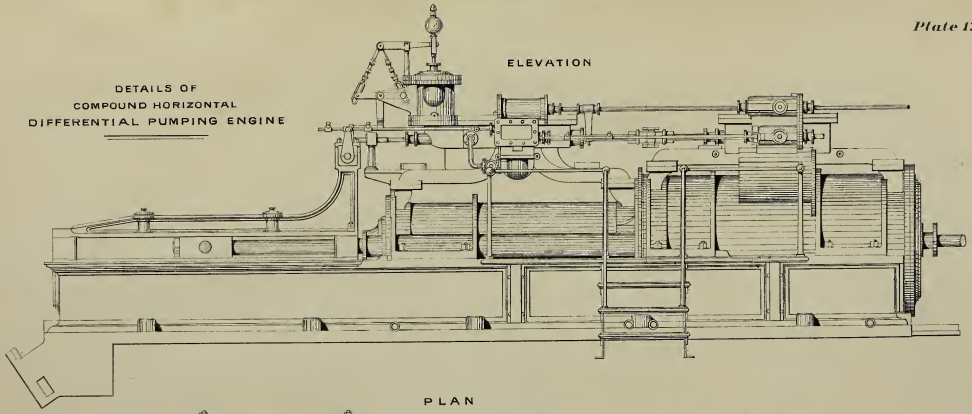


CROSS SECTION
THROUGH
LOW PRESSURE CYLINDER

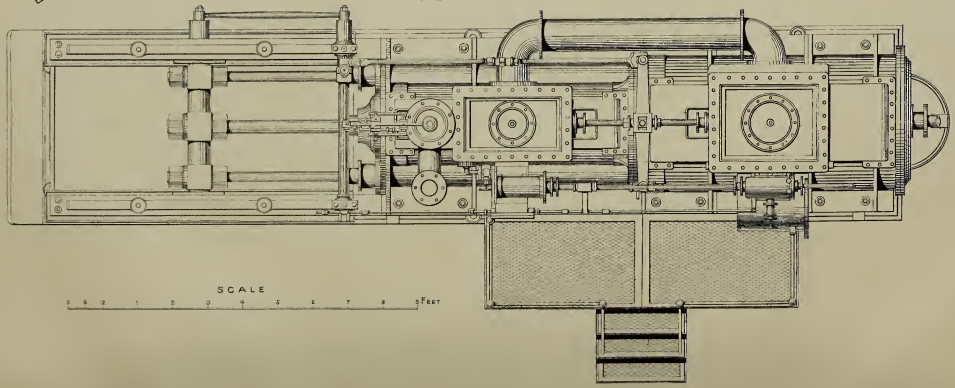


DETAILS OF
COMPOUND HORIZONTAL
DIFFERENTIAL PUMPING ENGINE

ELEVATION



PLAN

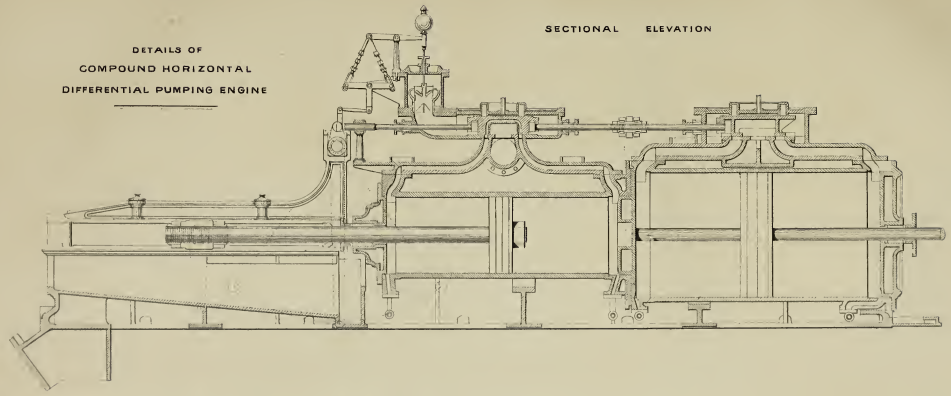


SCALE

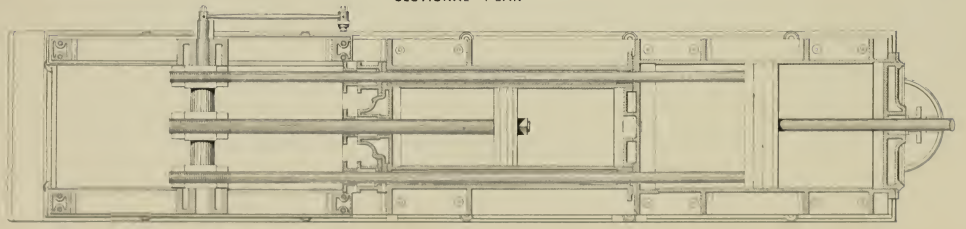
0 1 2 3 4 5 6 7 8 9 Feet

DETAILS OF
COMPOUND HORIZONTAL
DIFFERENTIAL PUMPING ENGINE

SECTIONAL ELEVATION



SECTIONAL PLAN



SCALE

Plates 10, 11, 12 and 13 were engraved from the drawings of the compound differential horizontal pumping engine, just completed, from our designs, by the Risdon Iron Works, for the Alta Mining Company. Plate 10 is a general side elevation and plan. Plate 11 shows two cross sections, one through the low and one through the high pressure cylinder. Plate 12 shows a complete side elevation and plan of the engine. Plate 13 shows a sectional elevation and sectional plan of the engine.

These illustrations are so complete that little description is needed. The back end of the high pressure cylinder forms the front cover of the low pressure cylinder, as shown in the sectional plan. There are two piston rods to the low pressure piston, which pass through tubes cast on the sides of the high pressure cylinder, as shown in plan and cross section. These two rods and that of the high pressure cylinder between them are all fixed to one cross-head, to which is attached the connecting rod for working the pump bobs. The cylinders are bolted down upon a massive bed-plate, the front end of which forms the guides for the cross-head. The condenser and air pump are behind the low pressure cylinder—the air pump being worked by a tail rod from the low pressure piston.

In this engine, the main steam valves, which are balanced D slides, receive their motion from a lever to the centre of which they are connected. This lever receives two motions. One end derives its motion from the main moving parts of the engine by another lever of the first order, which, receiving the full motion of the piston at its long end, imparts from its short limb the amount of motion suitable for the working of the valve.

The other end of the lever derives its motion from a small subsidiary piston, which receives its motion from the steam by means of a small slide valve. This motion is controlled by a cataract piston working in a cylinder filled with water or other fluid, which escapes from side to side through small openings that can be regulated at pleasure.

The motion of the little slide valve of the subsidiary cylinder is, in its turn, governed by the movement of another steam cylinder and cataract, which is shown in the drawings placed above the subsidiary cylinder.

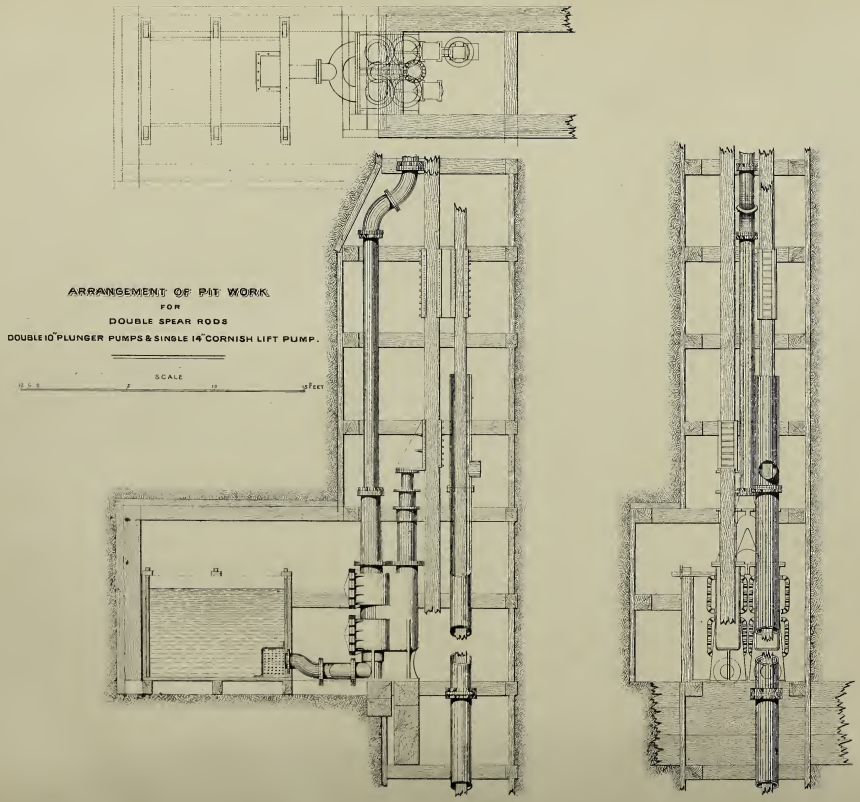
The action of this gear upon the motion of the engine will be best understood by an illustration.

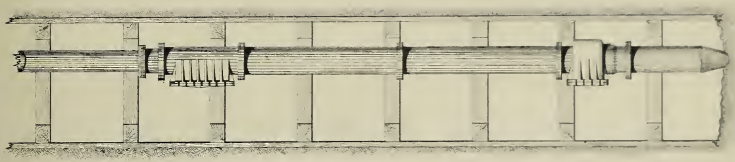
Suppose the main piston to be nearing the end of its stroke. The small arm on the left hand end of the rock shaft, best seen in the elevation, Plate 12, will engage with one of the tappets shown on the small valve rod of the upper or pausing cylinder, the tappet being so placed that the requisite

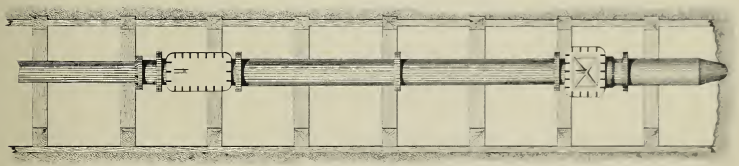
motion will be given to the small valve before the main engine comes to rest. Steam being thus admitted on the piston of the pausing cylinder, it will begin its stroke. Everything else being at rest, its rate of movement will depend on the amount of opening given to the cataract plug, which will be adjusted to the required amount of pause or rest at the end of each stroke. As the pausing cataract nears the end of its stroke, an arm attached to its piston rod (best seen on plan, Plate 12) engages with one of the tappets on the small valve rod of the subsidiary cylinder, and effects the required movement in this valve, admitting steam on the piston. Motion would thus be communicated through the main lever to the valves, and the main engine would commence its stroke as it moves. However, it is giving motion to the other end of the main lever, as already described, in a contrary direction to the motion communicated by the subsidiary piston, and cuts off the steam.

The main valve, therefore, has a differential motion, compounded of the motion derived from the direct action of the main piston and an opposite motion from the subsidiary piston. Now, the motion of the subsidiary piston is rendered constant by means of the cataract. Seeing, then, that the cataract end of the lever has a constant motion, independent of the engine itself, and that the other end must needs have a varying motion, depending on the varying load on the engine, then the resultant motion of the main valves being taken from the centre of the lever, and compounded of a varying and a constant motion, must also vary with every variation in the motion of the main piston. At the commencement of the stroke the cataract end of the lever has a lead in advance of that of the opposite end. With a constant load on the engine this lead will be constant, and the cut-off constant; but the slightest variation in the load causes a corresponding instant variation in the lead, and as a consequence the cut-off takes place earlier or later, as the load is decreased or increased. The engine thus automatically varies the expansion to suit the varying conditions of resistance. The pause at the end of each stroke is of great importance in a pumping engine, as it allows time for the valves of the pumps to fall by their own weight alone to their seats, preventing the possibility of a shock from sudden closing under pressure, and also preventing a loss of efficiency from slip.

The Compound Differential Engine is superior to the Cornish engine because, it being double acting, half the area of piston will do the same amount of work; with the two cylinders a greater expansion of steam can be realized, with the consequent economy of fuel; also, because it is safer


ARRANGEMENT OF PIT WORK
FOR
DOUBLE SPEAR RODS
DOUBLE 10" PLUNGER PUMPS & SINGLE 14" CORNISH LIFT PUMP.
SCALE
0 5 FEET 15





COMPOUND DIFFERENTIAL

HORIZONTAL PUMPING ENGINE,

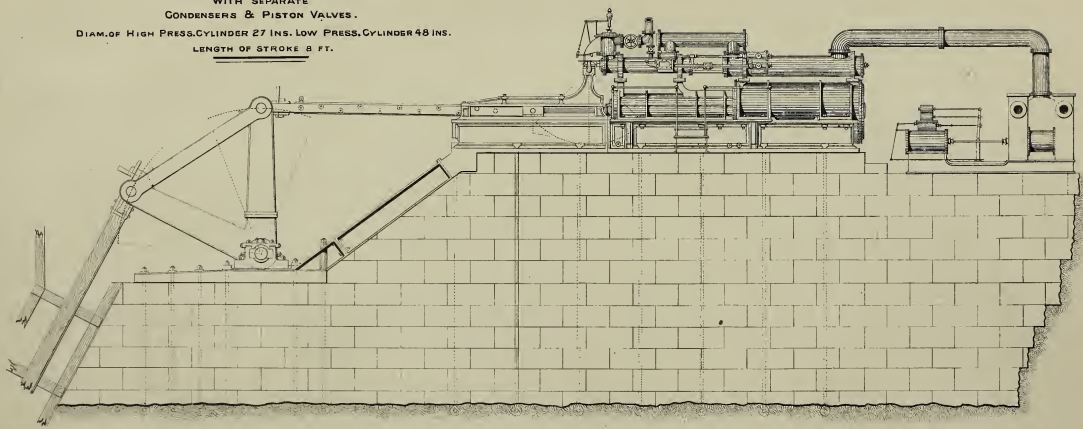
WITH SEPARATE

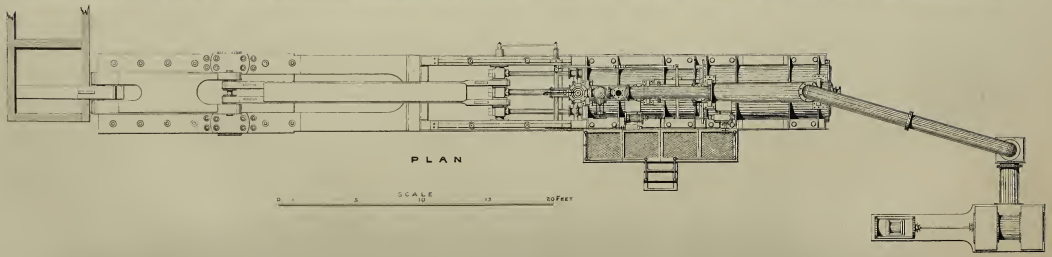
CONDENSERS & PISTON VALVES.

DIAM.OF HIGH PRESS.CYLINDER 27 INS. LOW PRESS.CYLINDER 48 INS.

LENGTH OF STROKE 8 FT.

ELEVATION.



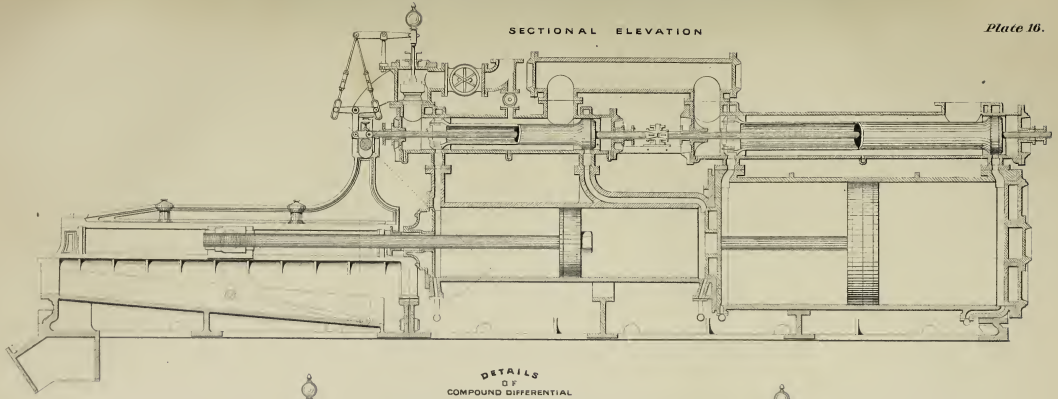


PLAN

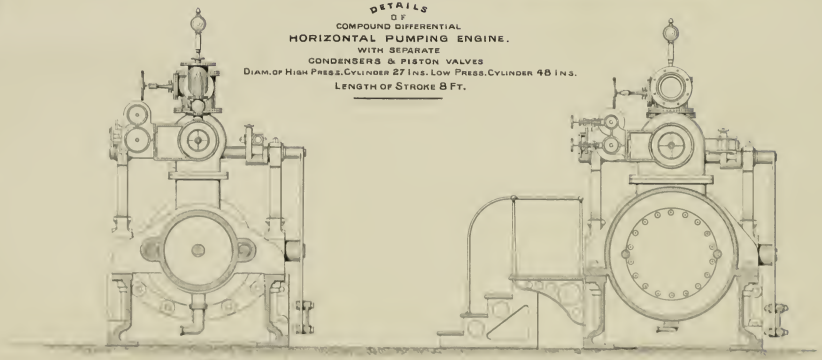
SCALE

0 5 10 15 20 FEET

SECTIONAL ELEVATION



DETAILS
OF
COMPOUND DIFFERENTIAL
HORIZONTAL PUMPING ENGINE.
WITH SEPARATE
CONDENSERS & PISTON VALVES
DIAM. OF HIGH PRESS. CYLINDER 27 INS. LOW PRESS. CYLINDER 48 INS.
LENGTH OF STROKE 8 FT.



CROSS SECTION THROUGH HIGH PRESSURE CYLINDER.

CROSS SECTION THROUGH LOW PRESSURE CYLINDER.

in working against variable loads, requires lighter pit work, and less costly buildings and foundation.

In the general plan and elevation of the Pumping Engine designed by us for the Alta Mining Company (Plate 10), it will be noticed that the pumps are actuated through two right-angle bobs, or bell cranks. This is the first introduction of this system to our quartz mines, although often used in England. Two strong cast-iron girders cross the shaft, as shown in the plan, in such a way as not to obstruct the shaft-opening. These girders are united to the engine bed-plate by a heavy cast-iron fork-piece, so that the whole forms one continuation of the engine frame, making it one machine, with all its strains within itself. The angle bobs are of cast-iron, in box form, cast in one piece, the ends of the arms being joined by heavy wrought-iron straps. A connecting rod ties the vertical arms of the bobs, which being always in compression is made of cast-iron. The bob furthest from the engine is fitted for a balance, the sinking or lift pump being attached to that bob. By this arrangement the one spear-rod balances the other; thus dispensing with balance bobs in the shaft, whilst the plunger pumps being double, discharging water both on the out and in stroke of the engine, the pump column does not require to be so large.

Plate 14 shows a plan and sectional elevation of the pumping compartment of the Alta shaft. This engine started with just the amount of pit-work shown in our engraving, consisting of one set of a pair of 10-inch plunger pumps, with a lift of 300 feet, and one 14 inch sinking Cornish lift pump, working a little over 200 feet. A second set of 10-inch plunger pumps are now being placed at the 600-foot station, the shaft having now reached that depth. This engine will be fully loaded when the shaft reaches a depth of 1500 feet. At the present time there is not much water in this mine, and the engine is only working three strokes in two minutes. The water will increase, however, as the sinking goes on. As soon as the second set of plunger pumps are attached, we will make some experiments with a view to find out the economy of this engine, and also to determine the value of the wood used as a fuel. The working of this engine, so far as we have yet been able to judge, leaves nothing to be desired.

Plate 15 shows a general plan and elevation of a Compound Differential Horizontal Pumping Engine, designed by us and under course of construction for the Meadow Valley Mining Company. This engine is to work a single line of 14-inch plunger pumps, the shaft being a straight incline 62° from the horizontal, and will start from a depth of 1400 feet on the incline. The dimensions of this engine are: Diameter of high pressure cylinder, 27

inches; diameter of low pressure cylinder, 48 inches; length of stroke, 8 feet. This engine is to have piston valves, details of which are shown in Plate 16, the object being to reduce the friction in order to place the engine more under the control of the cataract. We anticipate from this engine very good results. The angle bob, which, like those of the Alta, is of cast-iron in box form with heavy wrought-iron straps, but which, having to transmit the push as well as the pull of the engine, has a heavy cast-iron strut or abutment fitted and keyed between the straps, as shown. The center or rocker shaft of the bob is carried in heavy bearings resting on massive foundation plates, which are connected by heavy casting to the main bed of the engine; having in this way one continuous frame that embraces both the engine and pit-head work, relieves the foundation of most of the lateral strains, and on that account it need not be made so massive as it would require to be where the foundation is the only connection between the various parts. This is a matter of great importance when we consider the great cost of stone foundations in most of the mining regions of this country.

This engine is fitted with a separate or independent air pump, which is operated by a direct steam cylinder controlled by the differential valve gear. This plan has the advantage of being able to start with a vacuum, and it can be run for the hoisting engines, while the pumping engine is not at work.

The Table of Sizes here given for the Compound Differential Pumping Engine, will be found useful by those contemplating the erection of such machinery. It is calculated for a boiler pressure of 90 lbs. above the atmosphere, and as it sometimes happens that through movements in the shaft, and crushing of the strata, the friction of the pit work is increased by being forced out of line, we would recommend that in using this table the next size above that given for a certain amount of work should be taken. The quantity of water given is for a single stroke of the engine. Thus for a 10-inch single pump, 5 feet stroke, the engine making ten double strokes per minute, the water delivered would be 200 gallons.

TABLE OF SIZES AND CAPACITY PREPARED FOR THE COMPOUND DIFFERENTIAL PUMPING ENGINE.

By G. W. Dickie.

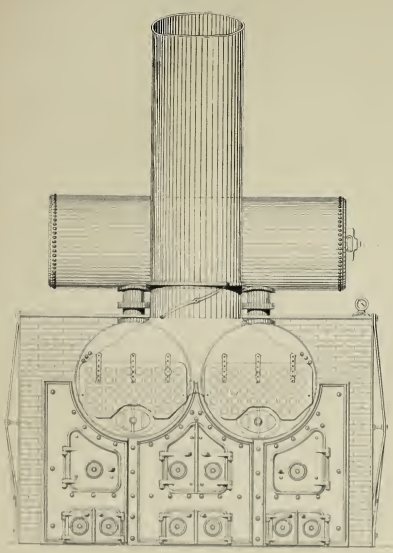
Diameter of Pump, Single.	Diameter of Pump, Double.	Gallons per Stroke of 3 feet.	Gallons per Stroke of 4 feet.	Gallons per St oke of 5 feet.	Gallons per Stroke of 6 feet.	Gallons per Stroke of 7 feet.	Gallons per Stroke of 8 feet.	Gallons per Stroke of 9 fe t.	Gallons per Stroke of 10 feet.	500 feet.		1,000 feet.		1,500 feet.		2,000 feet.		2,500 feet.		3,000 feet.		4,000 feet.	
										Dia. of High Pressure Cylinder.	Dia. of Low Pressure Cylinder.	Dia. of High Pressure Cylinder	Dia. of Low Pressure Cylinder.	Dia. of High Pressure Cylinder.	Dia. of Low Pressure Cylinder.	Dia. of High Pressure Cylinder.	Dia. of Low Pressure Cylinder.	Dia. of High Pressure Cylinder.	Dia. of Low Pressure Cylinder.	Dia. of High Pressure Cylinder.	Dia. of Low Pressure Cylinder.	Dia. of High Pressure Cylinder.	Dia. of Low Pressure Cylinder.
5¾″	4″	1.96	2.61	3.26	3.92	4.57	5.22	5.88	6.53	6 ″	10 ″	8 ″	14 ″	10 ″	17 ″	11½″	20 ″	13 ″	22 ″	14 ″	24 ″	16 ″	28 ″
7″	5″	3.06	4.08	5.1	6.12	7.14	8.16	9.18	10.2	7 ″	12 ″	10 ″	17 ″	12 ″	21 ″	14 ″	24 ″	15½″	27 ″	17 ″	29½″	20 ″	34 ″
8¼″	6″	4.41	5.87	7.34	8.81	10.3	11.7	13.2	14.7	8½″	14½″	12 ″	20½″	14½″	25 ″	17 ″	29 ″	18½″	32½″	20½″	35½″	24 ″	41 ″
10″	7″	6	8	10	12	14	16	18	20	9½″	17 ″	13½″	24 ″	16½″	28½″	19 ″	33 ″	21 ″	37 ″	23 ″	40 ″	27 ″	46½″
11¾″	8″	7.83	10.4	13.1	15.7	18.3	20.9	23.5	26.1	11 ″	19 ″	15½″	27 ″	19 ″	33 ″	22 ″	38 ″	24½″	42 ″	27 ″	46 ″	31 ″	53½″
13″	9″	9.91	13.2	16.5	19.8	23.1	26.4	29.7	33	12 ″	21 ″	17½″	30 ″	21 ″	37 ″	24½″	42½″	27½″	47½″	30 ″	52 ″	35 ″	60 ″
14½″	10″	12.2	16.3	20.4	24.5	28.6	32.6	36.7	40.8	14 ″	24 ″	19½″	33½″	23½″	41 ″	27 ″	48 ″	30½″	53 ″	33½″	58 ″	38½″	67 ″
15¾″	11″	14.8	19.7	24.7	29.6	34.5	39.5	44.4	49.4	15 ″	26 ″	21 ″	37 ″	26 ″	45 ″	30 ″	52 ″	33½″	58 ″	37 ″	64 ″	42½″	74 ″
17	12″	17.6	23.5	29.4	35.2	41.1	47	52.9	58.7	16½″	28½″	23 ″	40 ″	28½″	49 ″	33 ″	57 ″	37 ″	64 ″	40 ″	69 ″	46 ″	80 ″
18½″	13″	20.7	27.6	34.5	41.4	48 3	55.2	62.1	68.9	18 ″	31 ″	25 ″	43½″	31 ″	53 ″	35½″	61½″	40 ″	69 ″	43½″	75 ″	50 ″	87 ″
20 ″	14″	24	32	40	48	56	64	72	79.9	19 ″	33 ″	27 ″	47 ″	33 ″	57 ″	38 ″	66 ″	43 ″	74 ″	47 ″	81 ″	54 ″	93 ″
21¼″	15	27.5	36 7	45.9	55.1	64.3	73.4	82.6	91.8	20½″	35½″	29 ″	50 ″	35½″	61½″	41 ″	71 ″	46 ″	79 ″	50 ″	87 ″	58 ″	100 ″
22½″	16	31.3	41.8	52.2	62.7	73.1	83.6	94	104.4	22 ″	38 ″	31 ″	53 ″	38 ″	65 ″	44 ″	76 ″	49 ″	84 ″	53 ″	93 ″	62 ″	107 ″
24 ″	17	35.4	47.2	58.9	70.7	82.5	94 3	106.1	117.9	23 ″	40 ″	33 ″	57 ″	40 ″	70 ″	46 ″	80 ″	52 ″	90 ″	57 ″	99 ″	65 ″	114 ″
25¾″	18	39.6	52.9	66.1	79.3	92.5	105.7	119	132.2	25 ″	43 ″	35 ″	60 ″	43 ″	74 ″	49 ″	85 ″	55 ″	95 ″	60 ″	104 ″	70 ″	121 ″
27 ″	19	44.2	58.9	73.6	88.4	103.1	117.8	132.5	147.3	26 ″	45 ″	37 ″	63 ″	45 ″	78 ″	52 ″	90 ″	58 ″	101 ″	64 ″	110 ″	73 ″	127 ″
28¼″	20	48.9	65.3	81.6	97.9	114.2	130.5	146.9	163.2	27 ″	47 ″	39 ″	67 ″	47 ″	82 ″	55 ″	95 ″	61 ″	106 ″	67 ″	116 ″	77 ″	134 ″

Another style of pumping engine, which has obtained considerable popularity in the west of Scotland, is that patented and manufactured by Mr. Barclay, of Kilmarnock, Scotland. Our engraving (Plate 20), gives a side elevation of this engine, sufficiently clear to understand the leading features of construction. Mr. Barclay builds this engine as a rival of the regular Cornish-beam engine, and a comparison with Plate 1 will show at a glance what are the objects sought to be attained by this design. The points of superiority claimed for the Barclay engine are these, viz.: The maximum strains are greatly reduced by expanding in a second cylinder; the foundations are not so expensive; and the engine itself, by dispensing with the beam and substituting instead an overhanging lifting beam, is less costly. This engine is also built double acting, which is in its favor when compared with the single acting Cornish engine. We have made our illustration with the Davey motion attached, which would add an element of safety in working.

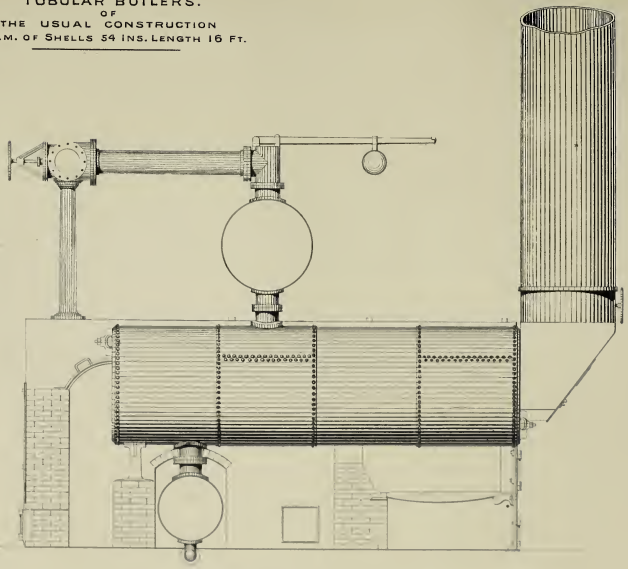
As this engine may be introduced to the notice of mining companies in this country, and as there are no Cornish pumping engines at work on our mines with which to compare it, we will have to compare it with such engines as that shown on Plates 12 and 13, or Plates 15 and 16. And for engines of moderate dimensions, such as we have shown, the Horizontal Compound can be built as cheap as the Barclay and will work as well, with the advantage of being able to work double spear-rods balanced, which we think is an advantage of some importance. For larger engines, the upright cylinders become a necessity, and we must compare the Barclay with such beam engines as we have illustrated on Plates 8 and 9. Here, in the matter of foundation (unless the nature of the ground favors the under-ground beam), the Barclay engine has the advantage. The lifting beam of the Barclay engine will also be cheaper built than the regular beam of the other engine; otherwise, the cost of either engine will be the same, the finish being equal. As to the relative economy of either engine there is nothing in the design of the one any more than in the other that can affect that question. Well loaded and properly worked, either the Barclay or the Beam Compound engine will give good results. It is saying a great deal for the Barclay engine when we state that it has become a formidable rival to the Cornish engine in situations where the latter has been in undisturbed possession for half a century. Let it come amongst us, and we will either beat it or admit its superiority.

In the matter of boilers no change, except in minor details, have been effected for several years. Nowhere has this important appendage of the

PAIR OF
TUBULAR BOILERS.
OF
THE USUAL CONSTRUCTION
DIAM. OF SHELLS 54 INS. LENGTH 16 FT.



FRONT ELEVATION.



SIDE ELEVATION.

SCALE

steam engine received that attention from engineers that the important part it plays in the general economy of the steam engine entitles it to. All our time has been devoted to devising changes, which are not always improvements in the motor that converts the steam into work, to the entire neglect of the generator that produces the steam. While it would be difficult in all our mining machinery to find two engines in all respects alike, it would be just as difficult to find two boilers that differed. Still we have no doubt that the boilers which generate the steam for our mining machinery are as economical as any other style of land boiler; nor do we at present see where any important improvement could be effected. Plate 4 gives a front view and sectional side elevation of a pair of 54-inch boilers, as usually set up. The style is so well known that our drawing needs no description.

Duty of Pumping Engines.

A cylinder of spring water, 1 inch diameter and 1 foot long, weighs ·341 lbs.

L=length of stroke actual.

N=number of strokes in one month.

H=height of lift in feet.

d=diameter of pumps.

q=number of cords of wood consumed in one month.

Duty=number of lbs. lifted 1 foot high per cord of wood.

$$\text{Duty}=\frac{[(\text{H}\times(d^2)\times\cdot341)\times\text{N}\times\text{L}]}{q}$$

The calculation has always been made for fathoms lift and for bushels of coal; but as we intend to try and get at some correct record of the duty of the various styles of pumping engines working at our own mines, we thought it better to adapt the formula to the conditions under which they work.

The usual mode of expressing the efficiency of a steam engine is in terms of fuel burnt per horse power, or units of work per hour. But as the efficiency of the boiler becomes in such a case mixed up with that of the engine, it is important that some method should be adopted by which the efficiency of either might be readily ascertained.

The efficiency of the boiler may be expressed in lbs. of water evaporated per cord of wood or per lbs. of coal, if that fuel is used; whilst the efficiency of the engine should be denoted in units of work done per 1 lb. of steam used. A formula for that purpose may be constructed thus:

Let 28=number of cubic inches of water in 1 lb. of steam.
I=initial pressure of steam.
S=its specific volume.
R=ratio of expansion employed.
a=average pressure per square inch=$\frac{(1+\text{Hyp Log R})\ \text{I.}}{\text{R}}$
U=units of work done by 1 lb. of steam.

Then $\text{U}=a\ \frac{(28\times 8\times \text{R})}{12}=$

The following table shows, in units of work, the comparative values of different degrees of expansion with a constant initial pressure equal to 100 lbs. per square inch.

Initial Pressure.	Ratio of Expansion = R.	Units of Work.	Average pressure. $=\frac{(1+\text{Hyp Log R})\ \text{I.}}{\text{R.}}$
100lbs.	0	63000	100lbs.
"	1.25	77017.5	97·8
"	1·66	94958·64	90·8
"	2·00	106596	84·6
"	3·00	132111	69·9
"	4·00	150192	59·6
"	5·00	164115	52·1
"	8·00	193536	38·4
"	10·00	207900	33·0

The formula for the duty of pumping engines would then become—
Duty = $\frac{(\text{H}\times(d_2)\times\cdot 311)\times\text{N}\times\text{L.}}{\text{W}}$ in which W = weight of water in pounds, and duty = units of work per pound of steam.

For the convenience of readily and accurately recording the duty of pumping engines, the water should be measured into the boiler by a meter made to register in pounds; and if at the same time the quantity of fuel burned were noted, the duty of both engine and boiler could be ascertained. The duty of the boiler would be found by dividing the number of pounds of water evaporated by the number of cords of wood burned, and the results obtained should then be expressed thus:

Duty of engine = number of units of work per pound of water evaporated.
Duty of boiler = number of pounds of water evaporated per cord of wood burned.

Such a record kept by the superintendent or engineer of each mine would be of great value in determining the relative efficiency of the various styles of pumping engines; also the efficiency of the class of boiler used as a steam generator. It would also fix the relative value of the pine wood used as fuel as compared with any known standard. Wherever practicable, the boilers for the pumping engine should be kept independent of the boilers for the hoisting machinery, as sometimes the reputation of a good pumping engine may suffer by receiving steam from the same boilers that supply defective hoisting engines.

Saturated Steam.			Hyperbolic Logarithms.			Properties of Steam.			
Absolute pressure in lbs. per square inch.	Temperature of Steam. Fah.	Specific volume.	Ratio of expansion = R.	Hyp. Log.	Portion of stroke at which steam is cut off.	Pressure of steam in lbs.	Latent heat of steam.	Sum of latent and sensible heat.	Temperature Fah.
5	162·3	4527	1·25	·223	$\frac{8}{10}$	1	1042·2	1140·2	104°
10	193·3	2358	1·43	·358	$\frac{7}{10}$	2	1029·6	1151·6	120°
14·7	212·0	1642	1·66	·506	$\frac{6}{10}$	3	1017·0	1157·0	140°
20	228·0	1229	2·00	·693	$\frac{1}{2}$	4·5	1004·4	1162·4	158°
23	235·5	1075	2·50	·916	$\frac{4}{10}$	7	991·8	1167 8	170°
30	250·4	838	3·33	1·202	$\frac{3}{10}$	10·	979 2	1173·2	194°
35	259 3	726	4·00	1·386	$\frac{1}{4}$	14 706	966·6	1178·6	212°
40	267·3	640	5·00	1·609	$\frac{1}{5}$	21	952·2	1182·2	230°
45	274 4	572	6·00	1·791	$\frac{1}{6}$	29	939·6	1187·6	248°
50	281·0	518	7·00	1·945	$\frac{1}{7}$	39	927·0	1193 0	266°
55	287·1	474	8·00	2·079	$\frac{1}{8}$	53	914·4	1198·4	284°
60	292·7	437	9·00	2·197	$\frac{1}{9}$	69	901·8	1203·8	302°
65	298·0	405	10·00	2·302	$\frac{1}{10}$	90	889·2	1209 2	320°
70	302·9	378				115	874 8	1212·8	338°
75	307·5	353	The above table of Hyperbolic Logarithms is given for the convenience of calculating the average pressure for a given initial pressure and ratio of expansion.			146	862·2	1218·2	356°
80	312·0	333				182	849·6	1223 6	374°
85	316 1	314				229	835·2	1227·2	392°
90	320·3	298	R = ratio of expansion.			279	822·6	1232·6	410
95	324 1	283	I = initial pressure.			348	808 2	1236·2	428°
100	327·9	270	A = average pressure. Throughout the stroke— $A = \frac{(1 + \text{Hyp. Log. R}) \text{ I}}{\text{R.}}$			402	795 6	1241·6	446°

Of course the average pressure in an engine at work should be determined by indicator card, especially so in compound engines.

Condensing Water.

As the temperature of the water from the mines, where that is used for the condenser, varies greatly, it would be well to use this formula:

W = weight of steam to be condensed.
T = its temperature.
l = its latent heat.
W'' = weight of injection water required.
t = its temperature.
t' = temperature of the mixture or hot well.
$W' = \frac{(T+l-t')W}{t'-t}$ For values of l see table of properties of steam.

Pumps and Pitwork.

It is not only desirable to have an economical engine, but it is also most important to use the power obtained from the engine economically. In other words, the engine should be coupled to an efficient set of pumps. Therefore in pumping from great depths the arrangement of the pumps and pitwork requires special consideration. The term pitwork is understood to include the whole of the apparatus and machinery beyond the outer end of the engine beam in beam engines, or below the nose of the working bob in horizontal engines, which is connected with the pumps.

In Cornish practice they use a very simple rule for calculating the quantity of water delivered from a given pump, as follows:

Let D = the diameter of the pump, then $\frac{D^2}{30}$ will represent the quantity of water in gallons delivered per foot stroke of pump. Let S = the speed of the plunger or bucket per minute, then $S\frac{D^2}{30}$ = the number of gallons delivered per minute. Let L = the lift in feet, and the number of horses' power exerted will be thus obtained. $\frac{L\times 10(S\frac{D^2}{30})}{33000}$ = the horses' power. This rule is for imperial gallons, and for standard gallons it would require to read thus: Gallons per foot stroke = $\frac{D^2}{25}$ gallons per minute = $S\frac{D^2}{25}$ horses' power = $\frac{L\times 8{\cdot}339(S\frac{D^2}{25})}{33000}$. No friction is allowed for in this rule for horses' power, so that whatever proportion of the indicated power of the engine this power is, will be the per cent. of efficiency.

The Cornish system of pit work is almost universally adopted in this country, sinking being carried on more or less in all the mines. The Cornish lift pump is found to be the best for the bottom lift—the other lifts being done by plunger pumps. The most important point to be observed

in plunger pumps is to have large, clear water ways, without sharp curves or square angles. Where no plant already in use has to be utilized, and with a vertical shaft, we think the double spear rod system the best—the delivery being continuous during both strokes of the engine, the pump column only requires to be half the area needed for a single plunger. The two spear rods balancing each other, no balance bobs and costly stations for them are required. The rods are not subjected to an alternate push and pull, as in the single rod system, but the strain being always in one direction, there is less risk of breakage and less guiding required for the spear rods. Plate 14, which shows the pitwork at present being fitted in the shaft of the Alta mine, is a very good illustration of the double spear rod system applied.

The first lift is a single 14-inch Cornish lift pump. Instead of the suction sliding down though a stuffing box, as is often the case in Cornish practice, the working barrel, as is usual in our mines, is made about 18 or 20 feet in length, and the whole is lowered as the shaft sinks. The pump column is of 15-inch lap welded tube, $\frac{1}{4}$ inch in thickness, through which an 8-inch spear rod works the bucket. The tank for the plunger pumps is placed so that the level of the water is above the suction valve. This is an important point in heavy pit work, for the security of the work depends particularly upon the complete charging of the pumps with water during the up or suction stroke, because at the end of that stroke, if the pumps are not charged solid, the weight of the pump spears carries the plungers back with a great shock. Stripping cocks should also be placed above the discharge valve, operated by a float in the tank, so that if the lift pump should be working on "fork," or not delivering its full capacity each stroke, the tank may be kept charged. The clack or valve chambers for each pump are cast in one piece with a part of the pump barrel, the chambers for each pump being bolted together by planed flanges, having one flat base resting on the foundation timbers, as shown. Two short pieces of pump barrel with the stuffing boxes are then bolted on the top, so that to put in a larger plunger only the short piece of the barrel has to be removed, the clack chamber and valves being made sufficiently large to admit a 12-inch plunger. The pump column is 10 inches diameter, of lap welded tube $\frac{1}{4}$ inch in thickness; each of the spear rods is 10 inches square; the connecting straps are 8 inches by 1 inch, and 16 feet long.

In Plate 3, we give a complete section of shaft drained by the spear rod system. Our engraving shows a shaft 800 feet deep, with a single line of 14-inch pumps, 8 feet stroke. The bottom lift is the usual Cornish sinking

pump, and the work above consists of three plunger lifts 220 feet apart. The water is discharged through a tunnel 46 feet from the surface. The balance bob shown at the 600 foot level is a cast and wrought iron bob of the latest style, introduced by the Risdon Iron Works. The column is all lap welded tube, with cast-iron flanges. This style of column has been used on all new works built during the past year, and obtains more favor than the riveted column used heretofore. The spear rods are 14 inches square, with four strapping plates on each joint or butt. These plates are 8 inches wide, $\frac{3}{4}$ inch thick, and 16 feet in length, with fourteen 1-inch bolts in each. As will be seen from our engraving, the plunger pumps used are of the ordinary description; in fact, we have simply reduced the drawings for a set of pit work now being built, and which differs little from what is found in the pumping compartment of any shaft on the Comstock.

In the single spear rod system plungers are sometimes substituted for balance bobs, and are when so employed constantly submitted to the pressure of the column of water in the pumps. They are fixed to the main rods precisely in the same way as the ordinary plunger; the objection to their use is a higher velocity in the pump column on the down stroke.

Amongst the improvements that have been effected from time to time in pumping machinery not the least were those effected in the pit work. The old Watt and Boulton pumping engines lifted the water on the steam stroke of the piston, by means of pit work composed of lift or bucket pumps. Murdock proposed, during Watt's time, to place in connection with the bucket or lift pump, the plunger (first proposed by Sir Samuel Morland), that the pumps might be made double acting, to suit the double acting engine. In the year 1796 one of these pumps was put down in the eastern part of what now forms the United Mines. The credit of first introducing it in its present form has been claimed for Trevithick, Davey, and Leon. To which it rightly belongs, it is difficult now to decide. We know, however, that from the year 1801 or 1802, it gradually came into use, and experience has demonstrated its great importance. The plunger pump was the invention of Sir Samuel Morland, who patented it in 1675, so that we are past the second centennial of the plunger. A similar kind of pump was, however, known to the ancients, an account of which is given by Hero, in his *Spiritalia*, only it does not seem to have had a stuffing-box. Perhaps they made better work in the olden time than us.

The plunger soon proved a great improvement on the bucket, in consequence of which the latter has ceased to be used, except for the convenience of sinking. With the bucket the friction is great and the leather wears

very fast, especially if the water contain sand or grit. It has also the great defect of having to be drawn every time anything requires to be done to it, causing considerable delay; while the pump packing of the plunger can be tightened at any time while in motion, and is also very durable. With the bucket lifts the balance bobs had to take up the weight of the spear rods, less the friction for the down stroke; but with the plunger only the surplus weight of the rods above the weight of the rising column has to be balanced (we are writing now of single acting Cornish engines). In geared or double acting direct pumping engines, working a single line of pumps, the balance requires to be half the weight of the rising column added to the surplus weight of rods. In our lithographs of pit work, it will be seen that the bottom lift in a set of pumps, according to our present practice, is a drawing lift; and the object of this is obvious, for it would be exceedingly difficult to continue sinking the shaft, and follow up the work with plunger pumps that require strong foundation timbers.

In order to economize the power in pumping as much as possible, the water which collects in the higher levels should not be allowed to fall to the bottom of the shaft, but should be conducted to the cistern of the next lift below, so that it often becomes necessary to make the plungers in the higher levels much larger than those in the lower ones, seeing that they not only have to lift the water taken up by the drawing lift, but also various additions from higher levels. In quartz mines, where sinking is going on all the time and the water to be handled rather an uncertain quantity, it is a matter of some difficulty to have the pumps at each station of the proper capacity, and water has often to be allowed to slip down to the lower levels.

In changing from a vertical to an incline shaft, angle bobs are used to change direction of motion, these are of various forms to suit the location and angle, all of them may be said to be a kind of hinged strut, sometimes, they are made to serve also as balance bobs. Formerly these bobs were made of wood with iron truss bars; now for all important works they are either made entirely of wrought iron or the arms or struts made of cast iron, with wrought iron straps, (see the angle bob at head of shaft, Plate 15); this style we think best as it is more rigid, and has no pins or bolts to work loose.

In most of the Comstock mines the water is of such a nature, that it destroys iron very rapidly, and every care should be taken to protect iron work from the action of the water. We think the iron work placed in these mines is not sufficiently painted. Pump columns should always be

dipped in asphaltum, and iron bobs, strapping plates, and pumps thoroughly protected by repeated coats of oil paint.

In running incline drifts, sinking winzes, or where the power of the main pumping engine is not sufficient to reach the full depth of the shaft, it is often necessary to employ pumping power, in addition to, and independent of the main pumping works, to pump the water from the drifts, or winzes, or from the bottom of the shaft to the tank, at the station of the first permanent pump set. Ordinary steam pumps that can readily be had in the market are generally used for that purpose; and as most of the leading mines use compressed air for operating rock drills, and ventilating blowers; these pumps are worked by compressed air. This system of working the auxillary pumps, notwithstanding the great waste of power attending it, has its advantages. The mines generally are very hot, and therefore, steam pipes however well protected are objectionable; besides the great length of pipe required to reach remote workings, would be a greater source of waste than that attending the working of compressed air. An air pump, if we may be allowed the expression, therefore is needed that will work the air expansively and yet be perfectly portable, needing no foundation or setting up.

Impressed with the importance of doing something to mitigate the great loss of power constantly going on in our mines, from working air as if it were a non-expanding body instead of a gas; we have for some time given this subject our earnest study, and have designed a pump (see Plate 6) which we think will meet all the requirements of pumping in drifts or winzes, or even as a sinking lift in connection with the main line of pumps, and at the same time be as portable as any of the ordinary style of steam pumps, and at the same time work the air with a four-fold expansion.

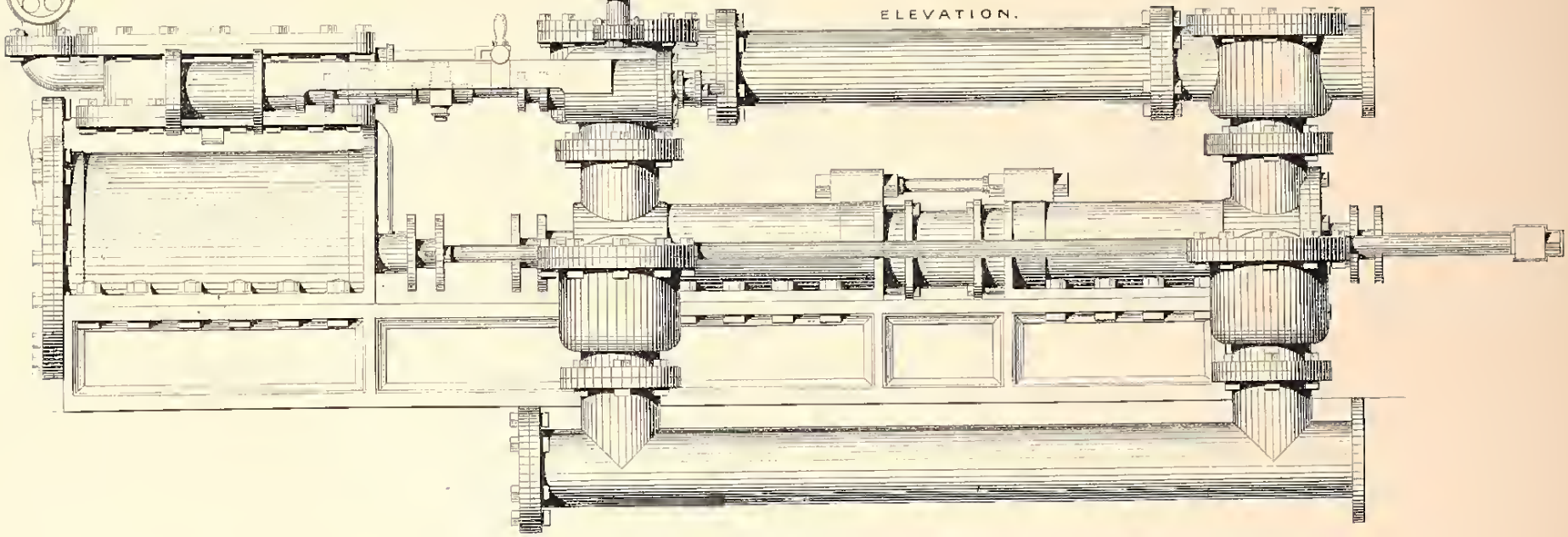
As will be seen from our engraving, the air cylinder is of the annular compound type, the high pressure cylinder being concentric with the low cylinder, the piston of which is an annular ring, the distribution of the air is effected by a double parted slide valve; the part in the valve forms the receiver between the high and low pressure cylinders; the motion of the valve is controlled by a modification of the Davey differential lever and cataract, the pump is of the double acting plunger type, and the valves, plain clacks with leather faces. Our drawing of the pump shows the arrangement so clear that no description is needed. Since our engraving was made we have modified the design somewhat, afraid that the compressed air holding, as it always does, some water in suspension by being expanded into four times the volume might, by the loss of temperature, cause the formation of ice in the passages leading to and from the low pressure cylinder. We

Plate 6.

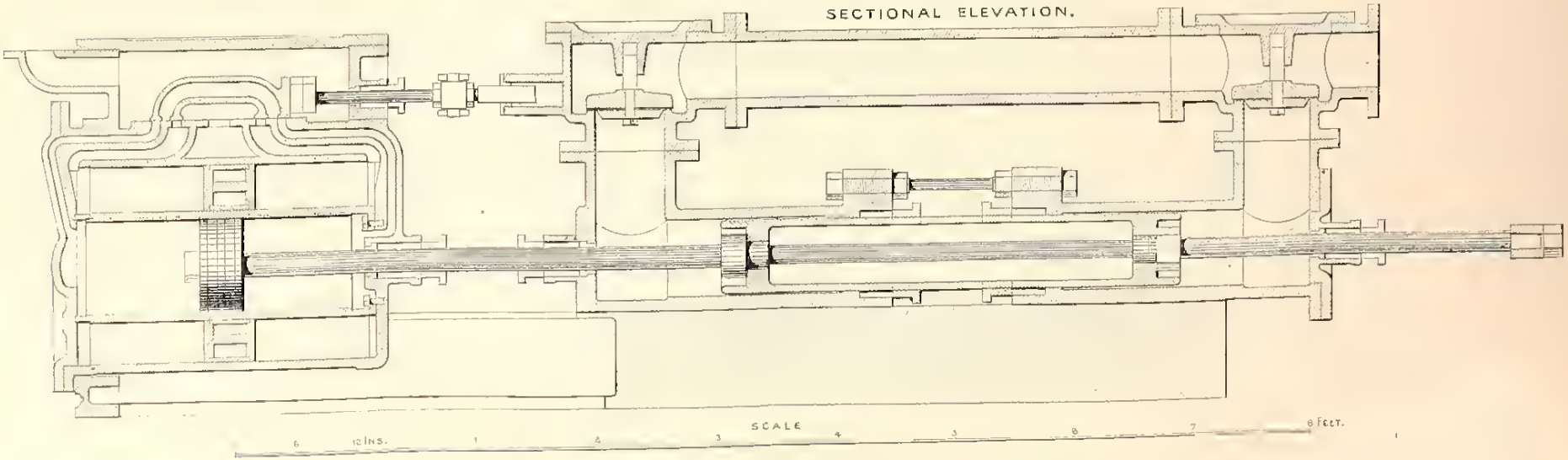
COMPOUND DIFFERENTIAL PUMPING ENGINE,
AND
DOUBLE PLUNGER PUMP.
TO BE WORKED WITH COMPRESSED AIR.

DESIGNED & BUILT BY THE
RISDON IRON WORKS S. F.

ELEVATION.



SECTIONAL ELEVATION.



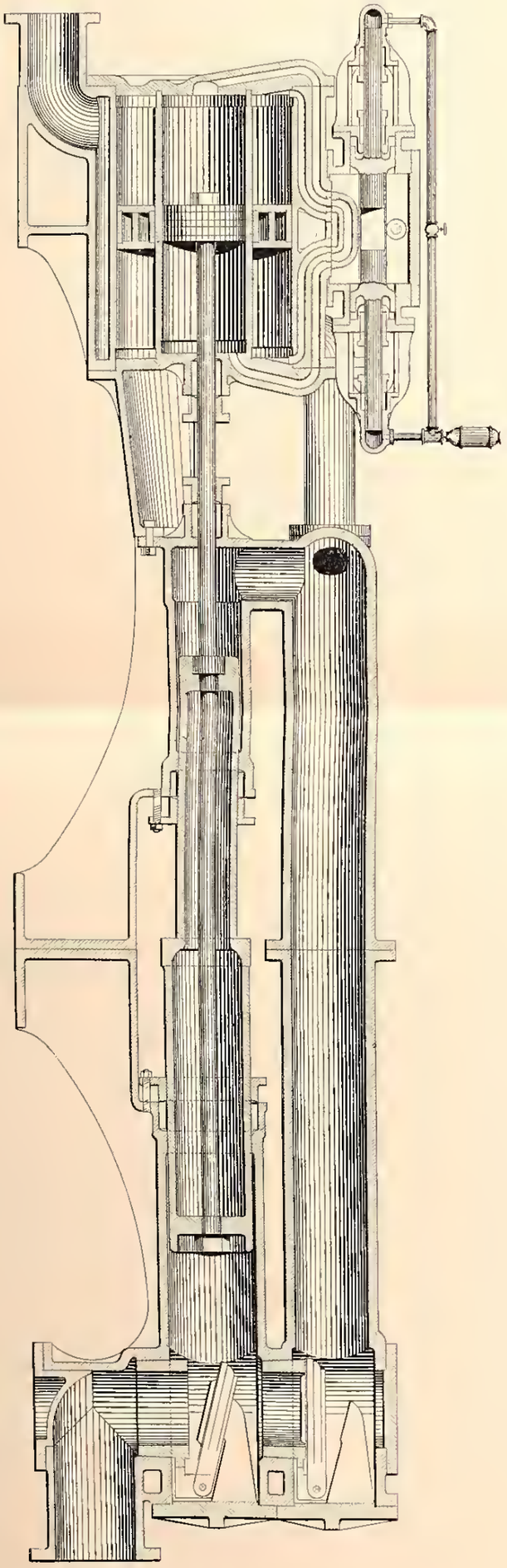
MOORE & DICKIE'S,

COMPOUND DIFFERENTIAL PLUNGER PUMPS.

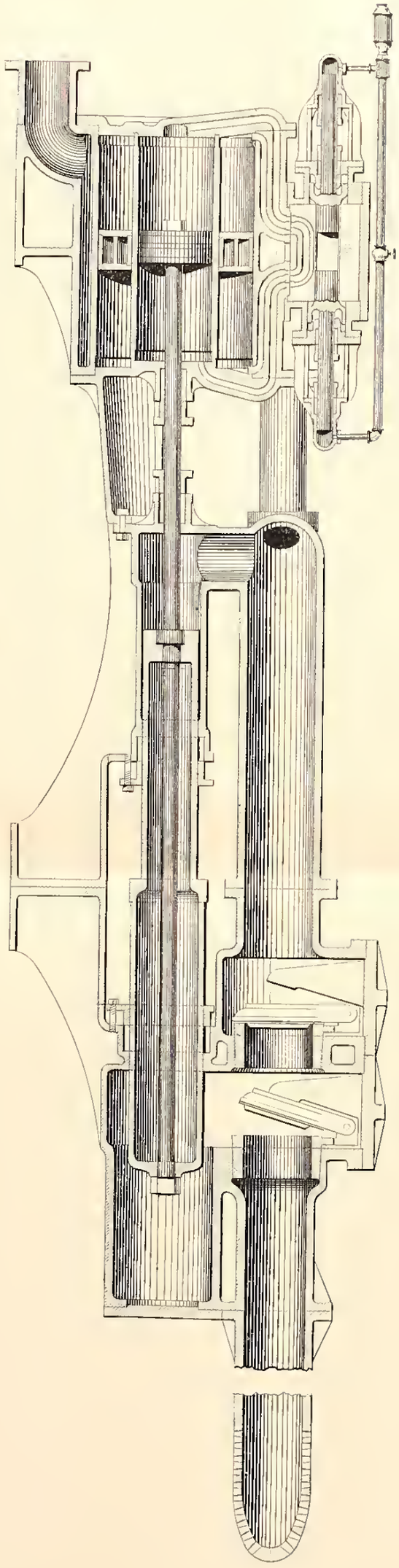
ARRANGED TO WORK WITH

COMPRESSED AIR.

SCALE, 3/4 IN. = 1 FT.



HORIZONTAL STYLE.



VERTICAL STYLE.

have taken advantage of the fact that the water in most of our mines is warm, and encased the cylinder in the discharge or column, or in other words, we have water-jacketed the low pressure cylinder and nozzles; should this pump be used for steam the discharge would then pass on without going into the jacket. For the sake of simplicity and compactness, we have discarded the differential lever, and after having it in many shapes have finally decided on the forms of pump shown in section on Plate 22. Our illustration shows the pump in two styles: one for working in drifts or incline shafts, where the incline is flat, and one for vertical shafts. It will be noticed that these pumps, though double acting, have only one suction and one delivery valve. The object of this is not so much for simplicity, although that important point is secured, as being able to keep the engine part of the pump further up from the bottom of the shaft or the sump. By this means, when placed in a vertical shaft, it works just like an ordinary Cornish sinking lift; in fact, it is a regular sinking pump, worked either by compressed air or steam. The principle of the differential ram used in this pump will be readily understood by an examination of our drawing, in our vertical pump: the suction for both strokes is accomplished during the up-stroke; the smaller, or top ram, which is just half the area of the bottom ram, at the same time discharges its volume into the pump column, and that quantity is delivered at the top. During the down-stroke the large ram delivers its volume through the discharge valve into the pump column; half of this quantity is delivered at the top, and the other half follows the small ram, thus balancing half the load on the large ram. This device is not new; but we think our application of it to sinking pumps, in a vertical shaft, for the purpose of getting more space between the engine and the bottom of the shaft, is a new application of the principle. It will be noticed that the valves used in these pumps are the ordinary clack-valves, as now used in plunger lifts, giving a clear, open water-way the full size of the column. We anticipate that this arrangement of pump will supply a want that has been long felt by mining superintendents. We know of some mines where ordinary steam pumps are used, worked with air to pump the water two hundred feet or so up to a point where the main pumping engine can reach it, where more fuel is required to compress the air for those pumps, than is required for the main pumping engine, working from fifteen to sixteen hundred feet of pumps; first, there is the loss in the compressor, then the loss by leakage in long lines of pipes, and last and greatest loss of all, working the air non-expansively in pumps defective, both in the engine, and water end.

Steam pumps the water ends of which have their valve chambers made up of a series of contracted and crooked water passages, should never be used in mining where the pump has often to be run fast, to get rid of water flows. From the report of pump test made at the Tenth Industrial Exhibition, held last year in San Francisco, the evils of contracted water passages is very apparent, even at a slow rate of speed. The suction and delivery pipes from these auxilliary pumps, like those of the main pumps, should have a clear area equal to that of the plungers; great waste of power is often the result of contracted pipes.

The pumping or compressing of the air for use in these auxillary pumps, and for drilling and ventilating purposes, has now become a distinct feature at all the principal mines, some of them employing as many as three or four large compressors. The Burleigh Air-compressor, which is built by the Burleigh Rock Drill Company, Fitchburgh, Mass., is a vertical machine, with one steam cylinder and two single acting air cylinders, set in line over a three-throw crank shaft, the cranks being set in the proper relative position to bring the points of greatest resistance in the compressors to coincide with the points of greatest effort in the steam cylinder. This machine has obtained considerable favor amongst the mines. The horizontal air-compressor, built by the Union Iron Works, San Francisco, is also employed to a considerable extent. It consists of an ordinary horizontal engine with variable cut-off, operating through a crank-shaft, a horizontal air-cylinder, usually of the same dimensions as the steam cylinder, the engine and compressor frames being of the same pattern, with a heavy fly-wheel between; the compressor cylinder and the piston are well water-jacketed. This machine has the advantage over the Burleigh, of delivering dry air, the Burleigh machine using a spray injection in the compressor cylinders. On Plate 7 we give elevations in view and in section of an air-compressor lately designed by us, from which we anticipate good results; one of 50-horse power is in course of construction which will be thoroughly tested. As will be seen from our engraving, it is a beam compressor, the steam cylinder being at one end and the compressor at the other end of the beam; the fly-wheels, of which there are two, are driven from a point near the cylinder end of the beam.

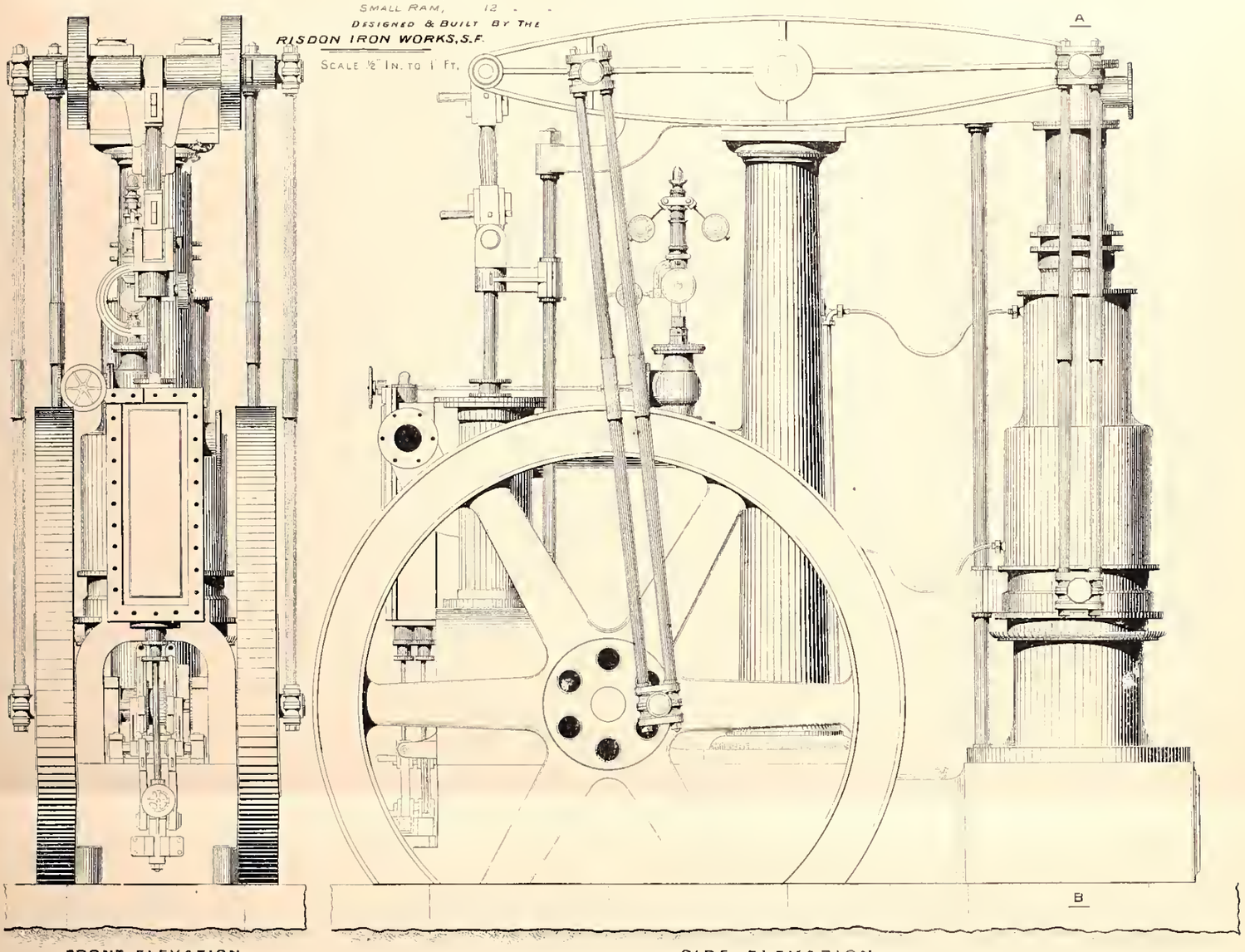
The compressor, as will be seen from the engraving, consists of two fixed rams, one of 26 inches in diameter secured to the bed plate, and another of 12 inches in diameter secured to the entablature; between these rams and upon them is a sliding cylinder water-jacketed all over; between the large

MOORE & DICKIES, AIR COMPRESSOR.

STEAM CYLINDER. 16 INS. DIAM. 30 INS. STROKE
LARGE RAM, 26¼ - -
SMALL RAM, 12 - -
DESIGNED & BUILT BY THE
RISDON IRON WORKS, S.F.

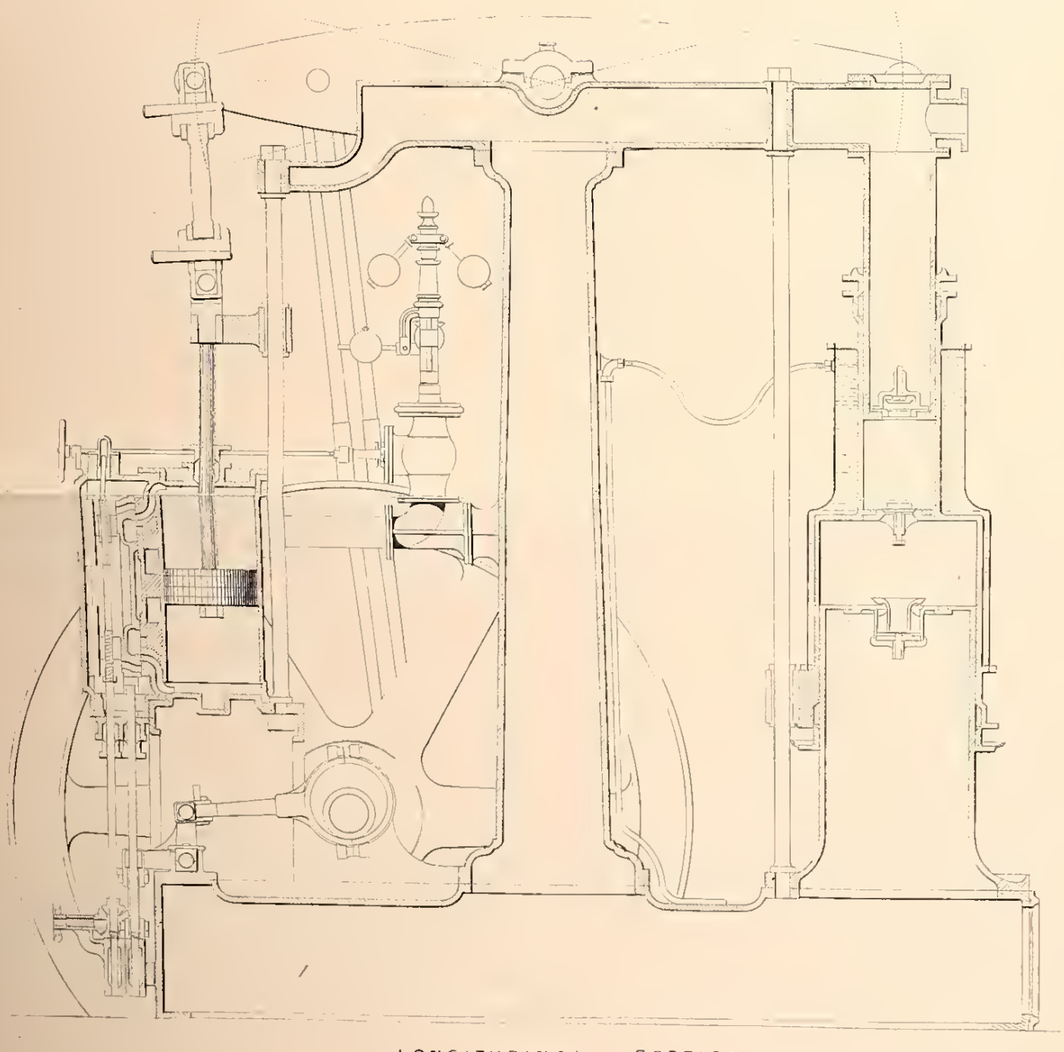
Plate 7.

SCALE ½ IN. TO 1 FT.

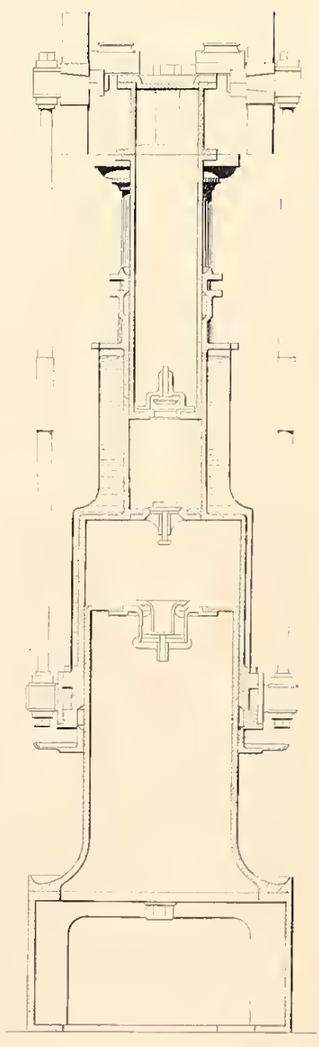


FRONT ELEVATION.

SIDE ELEVATION.



LONGITUDINAL SECTION.



SECTION THROUGH A.B.

and small end of the cylinder is a valve opening upward, and on the head of the large ram and the butt of the small ram are valves opening upward.

Suppose the sliding cylinder on the bottom centre and the steam cylinder piston at the top, steam being admitted on the piston, the sliding cylinder will rise and take air through the valve in the head of the large ram, until it reaches the top centre, the space above the large ram will then have air at the atmospheric pressure; steam being then admitted to the other side of the piston, the sliding cylinder descends, the valve in the head of the large ram closes, and the valve between the large and small cylinders opens, and the air during this stroke is compressed into the small cylinder; during the next up stroke this compressed air is forced through the valve in the bottom of the small ram into the receiver; at the same time the air is being taking again through the valve in the head of the large ram. The work of compression is thus extended over two strokes or a complete revolution of the engine, and more time is obtained for abstracting the heat generated in the operation; the valves which are only three in number, all lift upward and need no springs to seat them; all of the valves remain open during the greater part of the stroke, and have not such a quick action as when the compression is done in one cylinder, with the delivery valves only open for a small fraction of a second at the end of the stroke. We also consider the ram better than the piston, as any leakage is at once seen and readily stopped. The steam cylinder for this compressor is also fitted with a variable cut-off. In larger machines of this class we would make them double, using two compressor cylinders moving in opposite directions, and a compound condensing engine where water could be had.

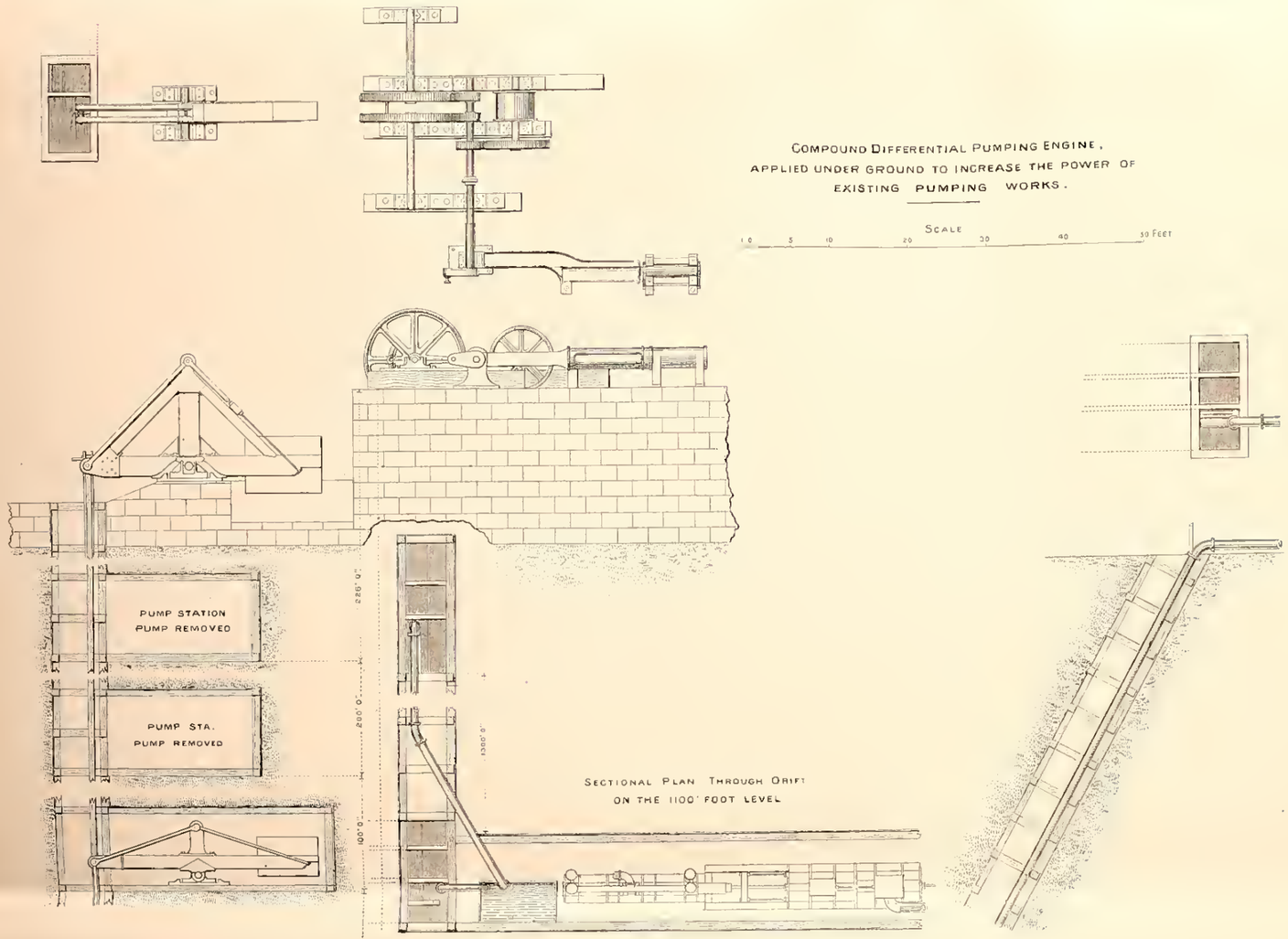
Compressed air is being used to such an extent in the mines, that anything that will tend to cheapen the compressing, will have its effect on the fuel bills. We therefore, feel excused in making this digression from what may be strictly termed, our subject.

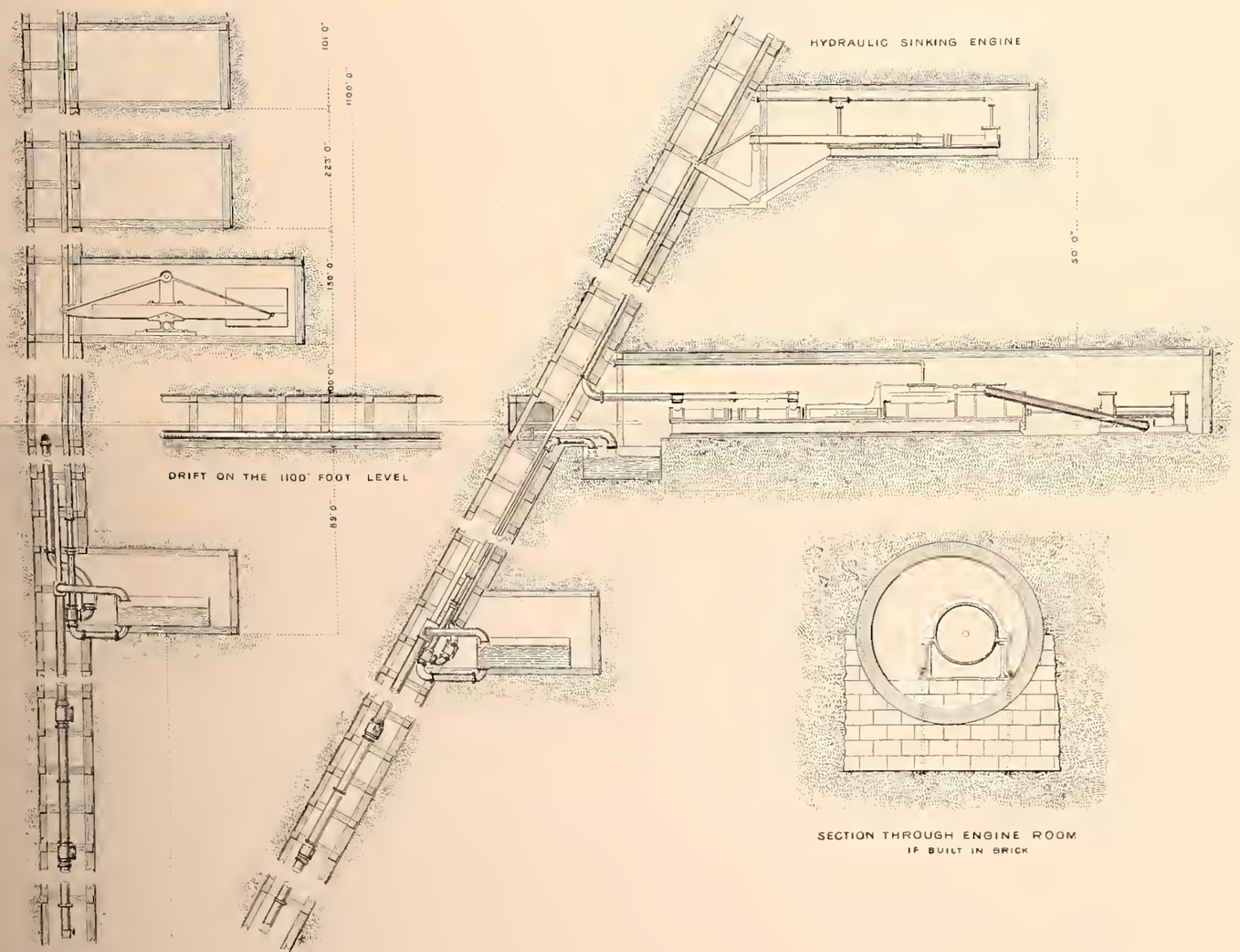
As in sinking, the drawing lift often works in fork, or in other words, has not enough water to fill the barrel, it is useful to be able to ascertain at any time the quantity of water which is being delivered. A convenient mode of doing it, is to lead the water delivered at the head of the column, to a small pond in which the velocity of the steam may be rendered almost nil, and to allow the water from the pond to fall over a weir 18 inches wide with vertical sides. A vertical post should be driven in the pond, back from the weir or sill, in still water, and on the post should be painted a gauge in inches and fractions, with zero at the level of the sill. The following table will then show the quantity of water passing over the weir:

WEIR TABLE FOR 18-INCH OVER-FALL.

Depth of Water. (Inches.)	Cube Feet per Minute.	Depth of Water. (Inches.)	Cube Feet per Minute.	Depth of Water. (Inches.)	Cube Feet per Minute.	Depth of Water. (Inches.)	Cube Feet per Minute.
1	7.725	7	142.974	13	361.296	19	639.216
1¼	10.730	7¼	150.637	13¼	372.587	19¼	652.445
1½	14.127	7½	158.55	13½	383.314	19½	665.122
1¾	17.893	7¾	167.04	13¾	393.866	19¾	678.03
2	21.770	8	174.627	14	404.296	20	690.168
2¼	26.016	8¼	183.	14¼	415.550	20¼	703.940
2½	29.490	8½	190.355	14½	426.495	20½	716.790
2¾	35.203	8¾	199.92	14¾	437.605	20¾	730.174
3	40.068	9	208.440	15	447.76	21	742.664
3¼	45.162	9¼	217.23	15¼	459.04	21¼	756.725
3½	50.488	9½	226.185	15½	471.575	21½	770.10
3¾	56.047	9¾	235.140	15¾	482.936	21¾	783.392
4	61.760	10	244.106	16	494.08	22	794.933
4¼	67.627	10¼	253.5	16¼	506.033	22¼	810.738
4½	73.648	10½	262.8	16½	517.729	22½	824.415
4¾	79.902	10¾	272 28	16¾	529.564	22¾	838.239
5	86.309	11	281.625	17	541.576	23	851.452
5¼	92.794	11¼	292.11	17¼	553.457	23¼	866.049
5½	99.610	11½	301.20	17½	565.470	23½	879.795
5¾	106.381	11¾	311.47	17¾	577.690	23¾	893.991
6	113.406	12	321.06	18	589.036	24	907.1
6¼	120.586	12¼	331.2	18¼	602.271	24¼	922.519
6½	127.920	12½	341.367	18½	614.601	24½	938.647
6¾	135.331	12¾	351.69	18¾	627.185	24¾	951.179

Having considered in a brief way the systems of pumping now in operation, and those about to be put into operation, we have now to deal with the most difficult part of our subject, viz., What system of pumping can be adopted that will meet all the difficulties that are ever present where heavy pumping works have to be constructed, and that will also meet the peculiar difficulties that have to be encountered in developing the great gold and silver bodies of ore that lie in the bosom of the Sierras. Our mining engineers have no fixed data to guide them in deciding for any given mine what amount of water he will have to contend with, at what levels he will strike water, nor does he know how deep he is going. In coal and iron mines it is generally known how far down the strata to be reached lies, and also from the record of other mines appear what amount of water will have to be lifted; thus permanent works can be erected that will work with a permanent load, and therefore under the most favorable conditions for economy.


COMPOUND DIFFERENTIAL PUMPING ENGINE,
APPLIED UNDER GROUND TO INCREASE THE POWER OF
EXISTING PUMPING WORKS.
SCALE
1 0 5 10 20 30 40 50 FEET
PUMP STATION
PUMP REMOVED
PUMP STA.
PUMP REMOVED
226' 0"
200' 0"
100' 0"
1100' 0"
SECTIONAL PLAN THROUGH DRIFT
ON THE 1100' FOOT LEVEL


101' 0"
1100' 0"
223' 0"
150' 0"
100' 0"
89' 0"
DRIFT ON THE 1100' FOOT LEVEL
HYDRAULIC SINKING ENGINE
50' 0"
SECTION THROUGH ENGINE ROOM
IF BUILT IN BRICK

Before entering into the consideration of any comprehensive scheme that would embrace any cluster of mines, such as the Virginia City mines, or the Gold Hill mines, let us consider some methods of increasing the capacity of existing works, without impairing the efficiency of either the old works or the new. In order to illustrate our meaning let us suppose a case or two, the counterparts of which will be readily recognized by those acquainted with the mines and their present water difficulties. We will first take the case illustrated by Plate 17; here are shown two shafts, one vertical and the other on an inclination of 62° from the horizontal. Suppose the incline shaft to have no pumping gear, the vertical shaft being fitted with a geared pumping engine of the most modern construction, working a line of 12-inch pumps, in five sets of plunger lifts and one draw lift. The two shafts are, say, 1300 feet apart; and on the 1100 foot level are connected by a drift, and sinking in each has been carried to a depth of 1300, or 200 feet below the drift. The strata is of such a nature that no pumping is required in the incline shaft as long as the vertical shaft is kept dry; but the water increases as the sinking proceeds and the pumping engine is run faster and faster, until breakages are of so frequent occurrence that sinking in both mines has to cease; the pumping engine being run at a speed that the gear and pit work will stand, and prospecting confined to the higher levels. Here then is the difficulty, the machinery will not stand the strains and shocks, resulting from the speed necessary to keep the mine dry. The pumps cannot be enlarged and run at a slower speed without the whole plant, except the engine alone being renewed, and even if that were done, sinking would soon stop for want of power.

To meet such a difficulty, we would propose to remove the four plunger lifts above the 1100-foot drift, as shown in the illustration; change the lower plunger to 15 inches diameter, and the suction left to the same size; let the pump column deliver into a sump on the 1100-foot level, place a compound differential pumping engine there to work a double-acting 17-inch plunger pump, delivering to the surface in one lift. This engine, of course, could be run slow or fast as the quantity of water required, or the pump column could be continued through the drift, which it might be in an open flume, to the incline shaft, and place the engine station there; then in case of any impervious strata occurring between the two shafts, the incline one could be drained by a hydraulic engine, taking its power from the pump column, and working a lift and plunger set of whatever size may be required. This would make the arrangement just as it is shown on the drawing. The hydraulic sinking engine could be run at any speed required

independent of the main engine; it would also be arranged so that its power would increase as the sinking progressed.

The objections that would be urged against this system are: first, the loss by radiation through a steam pipe 1200 feet long; and, second, having the main engine under ground, and out of sight. In stock companies, this second objection is of some importance, as a big show engine on the surface is an indication to the general public of a big mine below. The first objection we will endeavor to investigate. Mr. Davey, the patentee of the differential valve gear, in a paper read before the Institution of Mechanical Engineers, in London, 29th of October, 1874, gives some information on this subject, being the result of some experiments with a pair of differential pumping engines in the Morton Pit, at Clay Cross Colliery, near Chesterfield, England. These engines have a direct lift of 950 feet. The steam is conveyed down the shaft in a pipe $7\frac{1}{2}$ inches in diameter, and 1100 feet total length, which stands upon a timber bed at the bottom, and is stayed at every 27 feet length. There is a single expansion joint near the top, and the total expansion from the heat of the steam is 18 inches, amounting to nearly 2 inches per 100 feet in length. The whole length of pipe is clothed with non-conducting cement, and the steam is delivered into a receiver at the bottom 3 feet diameter and 6 feet long, in which the water formed by condensation is deposited.

The loss of steam from condensation has been ascertained by direct measurement of the water deposited in this receiver, and is found to amount to 8 cube feet of water per hour when the engines are standing, and 12 cube feet per hour when working at the ordinary speed of 10 double strokes per minute, with 45lbs of steam. Taking the evaporative duty of the coal at 9lbs per cube foot of water, the loss of coal from this condensation in the steam pipe amounts to 72 lbs per hour when the engines are standing, and 108lbs per hour when working; and the loss per 24 hours would therefore be 2160lbs if the engines are standing half the time, or 2592lbs if working continuously. We may state that the steam pipe was in the down cast compartment of the shaft, and consequently in a strong current of cold air.

From the results given in Mr. Davey's paper of the experiments at Clay Cross upon condensation in steam pipes, it appears that when the engines were working, the loss of fuel from condensation in the 1100 feet length of $7\frac{1}{2}$ inch steam pipe, was 108lbs of coal per hour; and as a $7\frac{1}{2}$ inch pipe would supply a 108 horse-power engine, working with little expansion and as within moderate limits, the condensation was not practically affected by the veloc-

ity of the steam in the pipes. We may safely conclude, that, for a single cylinder engine working non-expansively, and developing 100 horse-power, the condensation of steam in 1100 feet of pipe, 7½ inches diameter, would be found to be equivalent to about 1lb of coal per indicated horse-power per hour. But if a compound engine were being supplied from the same pipe, instead of an non-expansive engine as in the Clay Cross case, the power developed by the same weight of steam would be more than doubled; so, that for a compound engine working underground, 1100 feet below the surface, and having the steam carried down to it, the loss by condensation would be about ½lb of coal per indicated horse-power per hour, somewhere in the neighborhood of 15 per cent. loss. This is not a very great percentage of loss, and probably less than that occasioned by the friction of the massive spear rods, and other moving parts of the ordinary system of pumping. Our own experience enables us to prove the correctness of Mr. Davey's experiments. In the hydraulic machinery designed by us for the Palace Hotel, San Francisco, the accumulator and pumping engine are placed some distance from the Hotel, and the boilers; a 4-inch steam pipe is carried from the boilers to the pumping engines, a distance of 800 feet; this pipe is well covered with asbestos; there is a receiver in the accumulator house 20 inches in diameter and 6 feet high; the water by condensation is collected and drained off from this receiver. The cube inches of water collected for one hour equaled 5184 or 3 cube feet, the power developed by the pumping engines is 32 horses. Taking the same standard of value for fuel, 9lbs. of coal to evaporate one cube foot of water, the loss of fuel by condensation in steam pipe would be 27lbs. per hour, or a little over ¾ of a lb. per horse power per hour. We do not recommend this plan as shown on Plate 17, as the only or best plan for increasing the power of existing works, but simply as an indication of what might be done under certain circumstances; different methods of application of the same principle will suggest themselves to the minds of mining engineers and superintendents, as occasion or circumstances may direct.

Another method of working out the same principle for the purpose of dispensing with any further extension of the spear rod system, we will endeavor to illustrate by a supposed case, the counterpart of which may be found in Virginia city.

Suppose a vertical shaft sunk to a depth of 1700 feet, at the 1450 foot level an incline is started, which is intended to be the main shaft, the inclination is 37½° from the horizontal; fine new incline hoisting works have been erected with capacity of drum to wind 4000 feet of rope, which is supposed to be the ultimate extent of the shaft. We will say that this mine

has never been very wet and the pumps in the vertical shaft are only 11 inches in diameter; now the chances are in this case that water will be met with as the sinking of the incline progresses, the small pumps in the vertical shaft having enough to do in pumping the water collected from the various drifts above, cannot take any water that may be pumped from the incline; the company owning such a mine would therefore have to face the fact of new pumping plant being required for the whole mine.

Now in the particular case that we have in our mind at the present moment, the surface works are all new, having only been in operation for about one year, and consist of a geared, horizontal, high-pressure, expansive pumping engine, of the most modern construction; diameter of cylinder, 24 inches; length of stroke, 60 inches. The gear is similar to that shown on Plate 3, and consists of two spur pinions keyed on the engine shaft, 33.55 inches diameter at the pitch line, 5-inch pitch, and 14-inch face, 21 teeth in each pinion. These gear with a pair of massive spur wheels, 152.80 inches diameter at the pitch line, 5-inch pitch and 14-inch face, 96 teeth in each; the gear is, therefore, 4.571 to 1. The wrist-pin, for the connecting rod to the bob, passes through hubs in the arms of both wheels, and the longest stroke is 8 feet. Working with 6 strokes per minute for the pumps, which is fast enough for deep working, the piston speed of the engine is only 274 feet per minute, which is only 2.854 to 1 of the pump plungers. The present 11-inch pumps, making 6 strokes of 8 feet per minute, will deliver only 237 gallons without any slipage. Now it would not be safe in this case to count on less than 400 gallons per minute, and this would require a 14-inch pump making a little over 6 strokes per minute.

Now, let us see how far the present engine and gear would go with 14-inch pumps, working with 90 lbs. steam on the boilers; say 80 lbs. initial pressure in the cylinder, and the cut-off set at $\frac{1}{4}$ of the stroke. Area of cylinder = 452·39 square inches mean pressure = $\frac{(1 \times \text{Hyp. Log. of } 4 = 1{\cdot}386) \times 12}{4}$ = 54·878, less 14 for atmosphere and back pressure at the altitude of Virginia City, = 40·878 lbs. mean pressure, × 452·39 = 18492·8 lbs. pressure on the piston, the speed of the piston being 2·854 to 1 of the pump plungers, and the pump being single acting and the engine double, the pressure on the pump column would equal 18492·8 × 2·854 × 2 = 105557 lbs. From this we will have to deduct 30 per cent. for the friction of spear rods, transmissions, etc., making the effective pressure on plungers 73890 lbs. The area of a 14-inch plunger is 154 square inches; therefore, $\frac{73890}{154}$ = 480 lbs. per square inch nearly, which is equal to 1116 feet. Now it would be folly for this

company to extend their spear rods down the incline and incur the heavy expense of an angle bob, when even with 11-inch pumps they cannot go far; and if larger pumps are needed they cannot go at all, with the surface plant as at present arranged.

Here an objection might be raised that we have assumed too low a mean pressure in our cylinder; that the steam could be admitted for a longer portion of the stroke than $\frac{1}{4}$, and by that means more work got out of the engine. This might be so if the load on the engine were a constant quantity, but the engine makes four and a half revolutions to one of the spur wheel that actuates the pumps, consequently the power ranges from nothing to its maximum twice during each four and a half revolutions of the engine, and none of the geared pumping engines have been fitted with fly wheels heavy enough to admit of an equal amount of power being developed during each stroke of the piston. On Plate 5 we give an indicator diagram taken from a pumping engine of the same dimensions, and geared in the same manner as the one we have supposed in our case. This card represents seven consecutive revolutions of the engine taken just previous to its removal for want of power to sink any deeper. This engine with a cut-off at $\frac{5}{8}$ of the stroke and 90 ℔s steam, can only produce an average mean pressure of 52℔s per square inch, and that with a fly wheel of 40,000℔s. This actual case shows that we have not placed the average pressure too low.

Now let us see what can be done with this plant. The first defect we notice is the slow rate of piston speed, only 274 feet per minute, let this be increased to 500 feet per minute, which would give 50 revolutions for the engine. New gears would be made 278.5 inches diameter, same face and pitch as before, using the same pinions, making the gear 8.3 to 1. The relative speed of the engine piston and pump plungers would then be 5.2 to 1. Add a condenser to the engine, let the cut-off be the same as before, $\frac{1}{4}$ of the stroke, 80 ℔s. initial pressure in the cylinder, plus the atmosphere, 12 ℔s. equals 92 lbs. mean pressure, then $= \frac{(1 \times \text{Hyp. Log. } 4 = 1.386) \times 92}{4} = 54.$-878 less 2 ℔s. for imperfect vacuum = 52.878 ℔s. mean pressure × 452.39 = 23921.5 ℔s. pressure on the piston, the speed of the piston now being 5.2 to 1 of the pump plungers, and the pumps being single acting and the engine double acting, the pressure on the pump column would equal 23921.5 × 5.2 × 2 = 248783.6 ℔s., from which, as in the other case, we will deduct 30 per cent. for the friction of spear rods, transmissions, etc., making the effective pressure on the plungers equal 174148.5 ℔s., the area of a

14-inch plunger is 154 square inches, therefore $\frac{174,148.5}{154} = 1130.8$ lbs. per square inch, which is equal to a depth of 2630 feet. The incline starting at 1450 feet, and being on an angle of 37 degrees from the horizontal, can be run with this power a distance of 2480 feet. In these calculations we have supposed the pumping engine to be developing a constant power, but in practice this is not the case, as we have already shown. The engine making 8 revolutions for one of the pump gear, must either have a fly-wheel beyond all practical dimensions, or else the accelerated speed, while the pump plungers are at the ends of their strokes, will cause the governor to shut the steam off and the effective power of the engine is reduced; this is one of the defects of the geared system; in this case 12 times a minute the power of the engine varies from zero to its maximum and back to zero. Now, in order to be able to pump with this engine to the depth that our figures indicate, the gear must be so arranged that the power developed by the engine, shall be a constant, or near enough, for an ordinary fly-wheel to accumulate and give out the difference.

We will now endeavor to explain the method by which we propose to accomplish this result, and we might here observe that the plan we are about to propose, is applicable to all geared pumping engines that may be in the condition of our supposed case. We would propose in a case, such as we have described, to remove all the pit work from the vertical shaft, except what might be required to drain that portion extending below the head of the incline.

Put in a new line of 14-inch plunger pumps with the necessary pit work of the most modern construction. The sump or cistern at the head of the incline for the first plunger set to be made large to allow for slight variations in the action of the two systems. The two spur wheels which are now working as in Plate 3, to be removed and placed on the other side of the engine shaft, and geared with the pinions as before. New spur wheels twice the diameter of the present ones, to be made and geared with the pinions in the same way as before. Two heavy cast-iron frames would be made to carry all the pillow blocks, and extend out to the shaft, and carry the pillow blocks for the bob which would be raised to suit the centre of new wheels. By this arrangement, the old spur wheels would make two turns for one of the new, and by having them geared so that the wrist pin of the small wheel was up or down when the pin of the large wheel was on the horizontal or centre, they would always run in these relative positions. Now, by connecting the pin of the small wheel to the plunger of a hydraulic

pump of half the area of the 14-inch pumps in the shaft, working against an accumulate pressure of 623lbs, the power would be equally divided, and the hydraulic plunger being always in the middle of its stroke when the plunger pumps in the shaft were resting at the end of their stroke, the power developed in the engine cylinder would be a constant quantity. The accumulated power would be used to drain the incline by means of hydraulic pumps, as shown in Plate 18. The water delivered by the pumps of the hydraulic engines would go to the cistern at the head of the incline, but the water exhausted from the engine would return to the surface to be pumped again into the accumulator, thus securing pure water for transmitting the power. We think this the best plan for working geared engines to their full power; and seeing that there are so many fine geared pumping engines that cannot develop half the power they might do, for the want of some arrangement to make it a constant quantity, superintendents of mines, where this is the difficulty, would do well to study this system. What we have advanced may help them to prescribe for their own case. There are many different ways of applying hydraulics as an auxillary to the geared system. Almost every case requires to be treated on its own merits. In order to be safe from any mistake in its application, all the figures and quantities must be based upon correct data, or great loss may be the result. That it will be a help to many crippled pumping works at no distant day, we have not the slightest doubt.

We come now to the consideration of a comprehensive system of pumping, the application of which to the Comstock mines, has been our study for some time. To those who have watched the progress made within the past two or three years, in developing and prospecting of our deep quartz mines, the question must have often presented itself: "How deep is it possible to work these mines, and how far is it possible to work pumps through spear rods." The last question is the one which interests us just at present, and after a careful study of the subject in all its bearings, we have been forced to the conclusion that much of what has been lately done in pumping works is wrong; in some cases the question was, "is it advisable to use spear rods to work the pumps," when the question should have been "is it possible to use spear rods, and by them accomplish the object sought to be obtained." Pumping works have been designed and built recently for some of our deepest mines with such a reckless disregard of natural laws, that failures of the most discouraging nature must inevitably be the result, as soon as such plant may be called upon to do the work for which it was designed. Engines have been built to make a certain number of strokes

per minute, when by no possible amount of forcing can they be made to make half as many strokes. The fundamental fact appears to have been lost sight of, that the weight of the pit work to be put in motion and again brought to rest, every stroke that the engine makes fixes within very narrow limits the possible number of strokes that the engine can make. If we could approach this subject without compromising interests of great importance, and without appearing to attack the reputation of those interested, we would like to show here what is the maximum amount of work that can be done by some of the new pumping engines that have been lately started in the Comstock mines. As matters are we care not to disturb what cannot now be undone, but as we find much misapprehension amongst those whose opinion is sought on projected works of this nature, we feel it to be our duty to leave no doubt in the minds of those who may be guided by us, as to the conclusions to which we have arrived in regard to the possibility of a much further extension of the spear rod system of pumping in mines of great depth. Our purpose for the present we think, will be best served by a supposed case. Let it be required to drain a mine of say two thousand five hundred feet in depth, a single line of 18 inch pumps to be used, the shaft for 1500 feet is vertical, and the remaining 1000 feet is on an incline of 37° making 3160 feet total length of shaft.

If a compound differential double-acting beam engine be employed, having the same stroke of piston as the stroke of the pumps, the cylinders would require to be of the following dimensions, viz: high pressure cylinder $43\frac{1}{2}$ inches diameter ; low pressure cylinder 75 inches diameter. The length of stroke will be, say, 10 feet in both cylinders. These main points being determined we must now estimate the weight of pit work that requires to be set in motion and brought to rest again, for every stroke the engine may make. We will give our estimate of weights here in detail :

	Weight in lbs.
Engine pistons, piston rods, cross heads and connections	36,789
That portion of the beam that counts as moving mass	97,400
1500 feet of spear rod to the head of incline, 18″ x 18″=	162,000
Strapping plates for 35-foot lengths of timber, 344 strapping plates, 8″ x 1″ x 16′ =	144,480
3440 bolts, 20″ long, $1\frac{1}{8}$″, dia.², for strapping plates =	27,520
6 plungers for pumps in the vertical shaft =	32,346
Poles and connections to spear rods for pumps =	20,520
That portion of two balance bobs that counts as moving mass	67,890
Balance weights required for the bobs in vertical shaft	304,628
Strap connections to connect spear rods to beam	8,400
Carried forward	901,973

Brought forward	901,973
That part of the angle bob and connections that counts	53,000
1660 feet of spear rod in incline, average size, 14″ x 14″ =	108,336
Strapping plates for 35-foot lengths of timber, 282 straps, 6″ x $\frac{3}{4}$″ x 16′ =	67,680
2820 bolts, 16″ long, 1″ dia.² for strapping plates	14,100
5 plungers for pumps in the incline shaft	26,320
Poles and connections to spear rods for pumps	16,800
That portion of two balance bobs that counts as moving mass	48,400
Balance weights required for the bobs in vertical shaft	86,122
Guide wheels and axles for spear rod in incline	18,400
Total weight in motion during the up-stroke	1,341,131
3160 feet of 18″ water column =	344,932
Total weight in motion during the down-stroke	1,686,063

We think this estimate of weights rather under than over what would actually be used to work pumps of the size we have adopted. Area of pumps = 254 inches, and the actual head would be 2660 feet, allowing 100 feet for head lost at the various pump stations. The load on each side of the piston would = $\frac{2600 \times \cdot 43 \times 254}{2}$ = 141986 lbs., add 30% for friction = 184581.

On Plate 5 we have endeavored to produce the diagram of pressure on the two pistons combined at each point of the stroke, taking actual diagrams from engines of this class as a guide. We will now endeavor to follow the motion of the piston during the up and the down strokes of the pumps. The total weight set in motion during the up stroke, as we have shown, is 1,341,131 lbs, and the mean load on the pistons = 184,581 lbs. The mean pressure on the pistons during the first foot of the stroke = 237,150—184,581 = 52,569 stored up energy. This will make the velocity of piston at 1 foot = 1·584 feet per second. The mean pressure on the pistons during the second foot of the stroke = 246,784—184,581 = 62,203 + 52.569 = 114,772 lbs stored up energy. This will make the velocity of piston at 2 feet = 2·338 feet per second. The mean pressure on the piston during the third foot of the stroke = 238,663—184,581 = 54,082 + 114,772 = 168,854 lbs stored up energy. This will make the velocity of piston at 3 feet = 2·828 feet per second. The mean pressure on the pistons during the fourth foot of the stroke = 230,172—184,581 = 45,591 + 168,854 = 214,445 lbs stored up energy. This will make the velocity of piston at 4 feet = 3·162 feet per second. The mean pressure on the pistons during the fifth foot of the stroke = 222,789—184,581 = 38,208 + 214,445 = 252,653 lbs stored up energy. This will make the velocity of piston at 5 feet = 3·4 feet per

second. The mean pressure on the pistons during the sixth foot of the stroke = 204,884—184,581 = 20,303 + 252,653 = 272,956 lbs stored up energy. This will make the velocity of piston at 6 feet = 3·6 feet per second. The mean pressure on the pistons during the seventh foot of the stroke = 176,644—184,581 = 7937 lbs resistance + 272,966 = 265,019 lbs stored up energy. This will make the velocity of piston at 7 feet = 3·54 feet per second. The mean pressure on the pistons during the eighth foot of the stroke = 148,772 — 184,581 = 35,809 lbs resistance + 265,019 = 229,210 lbs stored up energy. This will make the velocity of piston at 8 feet = 3·3 feet per second. The mean pressure on the pistons during the ninth foot of the stroke = 102,442—184,581 = 82,139 lbs resistance + 229,210 = 147,-071 lbs stored up energy. This will make the velocity of piston at 9 feet = 2·64 feet per second. The mean pressure on the pistons during the tenth foot of the stroke = 36,916 — 184,581 = 147,665 lbs resistance + 147,071 = 594 lbs deficient in energy to complete the stroke which would thus be short of 10 feet by about $\frac{1}{4}$ of an inch. From these figures we obtain a mean velocity for the up stroke of 2·692 feet per second, equal to 3·960 seconds for the piston to move from end to end of its stroke, allowing a pause of only 2 seconds, would make 5·960 seconds for this stroke.

For the down stroke the mean load on the piston is the same as for the up stroke, viz: 184581lbs., but the weight set in motion is increased by the whole water column, and is as we have shown 1,686,063lbs., we will therefore follow it out as before:

The mean pressure on the pistons during the first foot of the stroke = 237150 — 184581 = 52569lbs. stored up energy, this will make the velocity of piston at 1 foot = 1.41 feet per second. The mean pressure on the piston during the second foot of the stroke = 246784 — 184581 = 62203 + 52569 = 114772lbs. stored up energy. This will make the velocity of the piston at 2 feet = 2.07 feet per second. The mean pressure on the piston during the third foot of the stroke = 238663 — 184581 = 54082 + 114772 = 168854lbs. stored up energy, this will make the velocity of piston at 3 feet = 2.52 feet per second. The mean pressure on the piston during the fourth foot of the stroke = 230172—184581 = 45591 + 168854 = 214445lbs. stored up energy. This will make the velocity of piston at 4 feet = 2.84 feet per second. The mean pressure on the piston during the fifth foot of the stroke = 222789 — 184581 = 38208 + 214445 = 252653lbs. stored up energy. This will make the velocity of piston at 5 feet = 3.08 feet per second. The mean pressure on the pistons during the sixth foot of the stroke = 204884 — 184581 = 20303 + 252653 = 272956lbs. stored up energy. This will make

the velocity of the piston =3.2 feet per second. The mean pressure on the piston during the seventh foot of the stroke = 176644 — 184581 = 7937lbs. resistance + 272956 = 265019lb. stored up energy. This will make the velocity of the piston at 7 feet = 3.16 feet per second. The mean pressure on the piston during the eighth foot of the stroke = 148772 — 184581 = 35809lbs. resistance + 265019 = 229210lbs. stored up energy. This will make the velocity of the piston at 8 feet = 2.82 feet per second. The mean pressure on the piston during the ninth foot of the stroke = 102442 — 184581 = 82139lbs. resistance + 229210 = 147071lbs. stored up energy. This will make the velocity of the piston at 9 feet = 2.36 feet per second. The mean pressure on the piston during the tenth foot of the stroke = 36916 — 184581 = 147665lbs. resistance + 147071 = 594lbs. deficient in energy to complete the stroke, which would thus be short of 10 feet by about $\frac{1}{4}$ of an inch. From these figures we obtain a mean velocity for the down stroke of 2.39 feet per second, equal to 4.14 seconds for the piston to move from end to end of the stroke; allowing a pause of only 2 seconds, would make 6.14 seconds for this stroke.

The up stroke, as we have seen, could be made in 5·960 seconds, which would give for the complete double stroke 12·1 second. The engine would, therefore, make 4·9 strokes per minute. But it must be remembered that in actual working the speed will fall short of what the figures give, and it would also be difficult to work such a large engine on two seconds pause; so that we would place the maximum speed at 4 strokes per minute. This would give, with 18-inch pumps and 10 feet stroke, 524 gallons per minute.

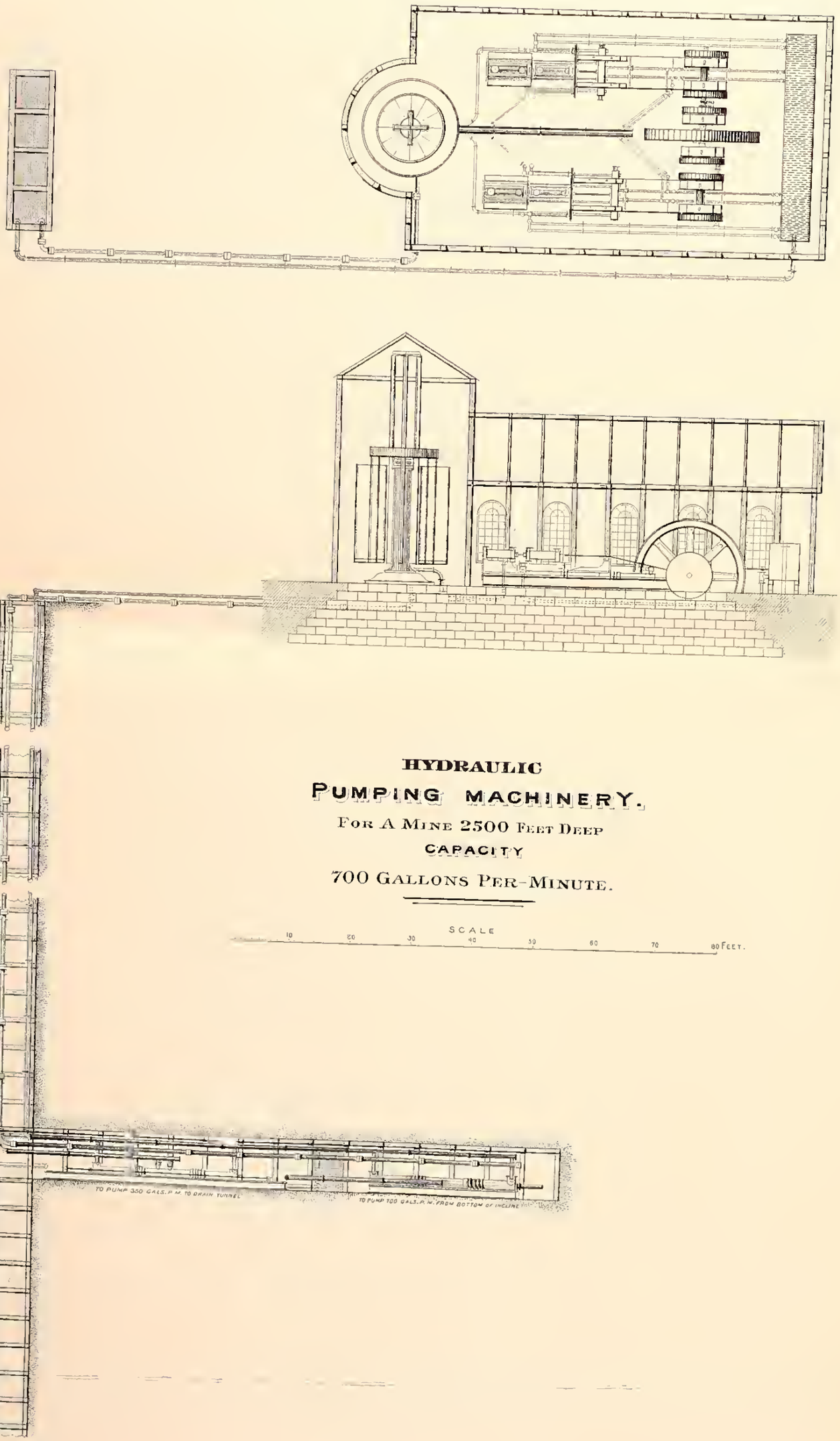
Thus we find that for deep workings, large pumps and large engines, working at a very low velocity, are required to pump a comparatively small quantity of water. This fact has, we think, been demonstrated with sufficient force by actual practice on our mines during the past year, to need no further illustration at our hands. Having felt the necessity of dispensing with the great moving masses required in the spear rod system, we were led to adopt hydraulic pumping as the plan most likely to be successful in draining great depths.

Hydraulic power has been in use for pumping for many years; in fact, we might say before steam. The hurdy-gurdy wheel, working a primitive lift pump, is pumping by hydraulic power. The principle is the same whether it be worked from a natural head of water or from an accumulator, and is that of employing water at a given head to raise a larger quantity of water against a less head. In hilly mining districts, water may be accumulated at a high elevation, and conducted to the mines in pipes, and there

employed in the differential, or other hydraulic engine, to pump the water from the mines. Thus the cost of steam power may often be saved.

In the system we are about to describe, we propose to accumulate the power of steam engines and transmit it through pipes to the pumping stations in the mine or mines to be drained. We have illustrated this system on Plate 18, in which we have shown a shaft 1400 feet vertical and an incline of 40° running to the 2500 foot level; supposing a pumping engine already to be at work on the mine, but of small capacity, say 350 gallons per minute from the head of the incline to the drain tunnel, and capacity being wanted for 700 gallons from 2500 feet, the hydraulic plant would have to pump 700 gallons to the head of the incline and 350 gallons from that point to the drain tunnel. Our engraving shows an engine house located some little distance from the shaft, clear of the other buildings, containing a pair of pumping engines; these are of the compound horizontal type, connected by a shaft and cranks at right angles with a heavy fly-wheel between. The high pressure cylinders are 27-inch and the low pressure 44, with a stroke of 6 feet. Each engine operates 4 hydraulic plunger pumps, $7\frac{1}{2}$ inches in diameter; these pump into an accumulator 26 inches diameter of ram and 16 feet stroke; the maximum speed of engines is 14 revolutions. From the accumulator, which is loaded for a pressure of 700 lbs. per square inch, the water is taken down the shaft in pipes to the main pumping station, shown at the head of the incline, and there operates the hydraulic pumping engines, returning again to the surface to be again pumped into the accumulator. The pumping engine, for the 350 gallons in the vertical shaft, draws its water from a large cistern built in the station, and has its engine and pump rams of equal areas, the valves being similar to those of the hydraulic pumping engine shown on Plate 19. The pumping engine for the incline, is placed in the same station and is larger; the pump portion of this engine has no suction or delivery valves, the ram simply moves the water column, which operates a plain pump ram down at the bottom of the incline; this plan makes only one station and removes the engines away from any danger of flooding and in some positions would be a necessity. This plan, however, only illustrates a part of our plans.

We would propose to group the mines and work them from a central station, and in order to illustrate our subject we will suppose the Virginia city mines to be drained in this way, and the depth to be worked to be 3000 feet, and the quantity of water that power would be required to be provided for would be, for the Consolidated Virginia & California, 500 gallons; the Ophir & Mexican, 400 gallons; Gould & Curry, 500 gallons; Savage,



HYDRAULIC
PUMPING MACHINERY.

FOR A MINE 2500 FEET DEEP

CAPACITY

700 GALLONS PER-MINUTE.

SCALE

10 20 30 40 50 60 70 80 FEET.

500 gallons; Chollar, 500 gallons; Hale & Norcross, 500 gallons; thus we have an aggregate of 2900 gallons to raise 3000 feet, or 72210000 foot lbs., equal to 2188 horses-power; and if we add 25% for transmission the total horses-power would be 2735. Of course this is greatly in excess of the pumping power now provided for these mines, but all this might be required before the 3000 foot level was reached, and the hydraulic pipes would have to be laid large enough to convey that amount of power; but the engine power need not be provided for the whole work until required. The central station or engine house, would be located in any convenient place, as near to the centre of the mines to be drained as possible, and made large enough for future additions to the engine power as it might be required. If we suppose that 1500 horses-power would be required within, say, 2 years after the completion of the works, then 4 sets of engines would be provided of 500 horses-power each, the spare set being provided in case of any repairs being done on any of the others. Each set of engines would be constructed after the very best type of compound condensing engines with one common separate condenser. Each engine would actuate a set of eight hydraulic plunger pumps, so arranged that any number can be thrown out of gear; all the pumps draw water from a common cistern, which receives the water that returns from the mines after doing its work, and pump it into accumulators, of which there would be four at the start, loaded to a pressure of 800 lbs. per square inch, equal to a head of water or column of 1860 feet; these accumulators would consist of a strong cast iron cylinder 16 inches internal diameter and say 25 feet long. The accumulator rams would be 16 inches diameter and about 28 feet long, having a dead load consisting of sand, stone, or scrap iron, heavy enough to secure the desired pressure; the weight boxes, in working up and down, would be guided in a suitable frame work; the accumulators would be all arranged to work together, and to stop and start the main engines, as the requirements of the work demanded. Each accumulator ram having an area of 201.0624 square inches, and the load, including ram and box, would be 160849.92 lbs., or a little over 80 tons, or the stored-up power in the four accumulators would be equal to 512 horses, or a little over $\frac{1}{3}$ of a minute full work, but the power required in pumping being always a constant quantity, variations would only be slight. Suppose one pump to be stopped in one of the mines of 25 horses-power and the accumulator half down, there would be ten minutes for the main engines to accommodate themselves to the alteration in power before the accumulators reached the top; then if the pump in the mine were started up again, the engines would have 20 minutes

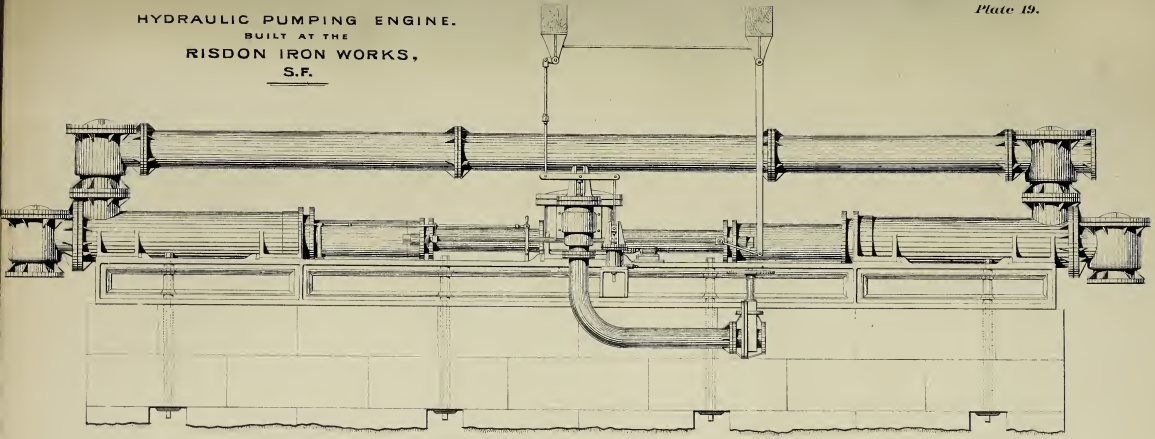
to accommodate themselves to the increase, so that the accumulated power here provided, is far in excess of that usually provided in the fly-wheel of rotating engines.

The engines and the boilers would be the very best of their kind that could be built; every radiating surface would be thoroughly protected by the best known non-conductor, so that we would anticipate obtaining 26 horses-power for each cord of wood burnt per day; then for 1500 horse-power, which is far in excess of the present pumping power required, there would be burnt, say, 60 cords of wood, a great saving over the present system. From the accumulator house the main pressure pipes are led, one going in each direction, these would be 12 inches in diameter and $\frac{9}{16}$ of an inch in thickness, of lap welded tube, jointed with flanges; a return pipe 14 inches in diameter of light iron would be laid along side of the pressure pipe to convey the exhaust water back to the cistern. From the main pressure pipe the power would be taken down the various shafts to the pumping stations below, these pipes would be of the requisite size to convey the amount of power likely at any time to be required. The pumping stations would be 600 feet apart; and the pumps, except the sinking lift, might be Davey's Patent Hydraulic Pumping Engine. We have adopted this style of pump, because it has been well tried and proved to be an excellent arrangement for underground work. Plate 19 is an elevation and section of this pumping engine. The novel point in the design is the method of applying the force of the water power directly on, or rather within the pump plunger, through a fixed hollow ram; this enables all the packing to be done by outside stuffing boxes, a very important point, especially in pumping dirty or gritty water, which is always the case in quartz mining. The valves are all operated directly by the water pressure and give a clear straight opening for the water.

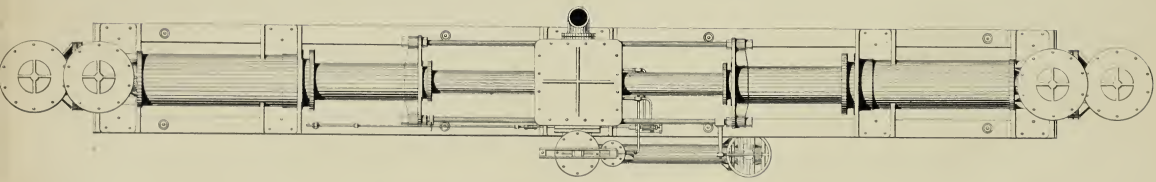
These engines being self-contained, like an ordinary steam pump, require little or no foundation, and can be set to work in a few hours after they are at the station. The valves are made of the best gun-metal, and are miter valves with large areas, producing no throttling, and will work under the heaviest pressure without excessive wear, and they are also under self-control by means of the differential gear.

The sinking hydraulic engine, to work the bottom lifts in the various shafts, might be the same as that shown on Plate 18; or it may consist of three simple single acting rams, the valves being similar and worked in the same way as those of the other engines, these rams lift a cross-head, to which the spear rods are attached, working an ordinary Cornish lift

HYDRAULIC PUMPING ENGINE.
BUILT AT THE
RISDON IRON WORKS,
S.F.



ELEVATION.



PLAN.

SCALE

pump. Each of the three rams are supposed to be equal to a lift of 100 feet, so that one, two or three rams can be used according to the length of the lifting column, small balance rams may also be added to adjust the weight of the spear rods to the work, 300 feet being the full capacity of this pump and there being 600 feet between the permanent pump stations an intermediate pumping engine of 300 feet capacity, would be used on the lower station until depth enough for a permanent station was reached. The useful effect in these hydraulic engines, is about 80 per cent. of the power applied, which is much higher than can be obtained by any spear rod system. In fact, the chief object in designing this system, is to be able to drain the mines without the delays incidental to the use of spear rods, and this advantage will be fully appreciated by mining engineers. We have already stated that the great difficulties the mining engineer has to meet in designing pumping works for our gold and silver mines, are those that arise out of the uncertainty, both as to quantity and depth. Now this system in providing the power at a central station for a number of adjacent mines, can throw all its surplus power on a minute's notice to any pumping station in any of the shafts, where a water reservoir may be struck. This feature is one of vast importance, for by the present system, if a sudden increase of water be struck and the surface works should have a capacity of 250 horses-power and already loaded to 200, there is only 50 horses-power that can be applied to meet it, may be an increase equal to or greater than the whole average load; in such a case the mine must be flooded up to the natural height that the water will rise, and every mining engineer knows what losses follow such an accident, and such cases have occurred but lately in some of the mines we have selected to illustrate this hydraulic scheme. Now, if this supposed case formed part of a hydraulic system of pumping, such as we have indicated, and using on an average 200 horses-power, the average power of the united mines being 1500 horses, and as we have provided 2000 in the accumulator house, there is a surplus of 500 horses-power that can be immediately set to work in the flooded shaft; the hydraulic engines for transmitting this power being all alike, fitted for 600 feet lifts, several of which could be kept in readiness at the central station to meet such an emergency; it would only be a work of two or three hours to double the pumping capacity of any given shaft. Sometimes it may be found necessary to pump local bodies of water from the stations in the higher levels as well as that from the station below. Where this difference is slight, the number of strokes made by the hydraulic engine at that station, may be increased to the required capacity, or if the local quantity be great,

an extra pump can be placed at that station without in any way affecting the working of the other pumps in the mine; these pumps having no working parts that can take hurt by immersion, will continue working under water as long as the packing lasts. Davey, of Leeds, mentions a similar pump built by his firm that continued working under water for three months, doing a duty of 80 per cent. We have studied this subject in all its bearings for the past two years. In the beginning of 1875, Mr. Moore proposed to the Belcher and Crown Point companies to drain their mines by a hydraulic system, careful plans and estimates for which we prepared at the time; we still think that plan would have given better results than the plan which was finally adopted. In fact, the hydraulic system we know from our own experience in hydraulic machinery, which has been quite extensive, is remarkably simple and free from trouble or danger, less really than attends the distribution of water through the pipes in a city.

If we compare this method with any of these now in use at our mines, it will be found to possess advantages in every important particular; it occupies no room at the shaft mouth, where room is so valuable and good foundations are difficult and costly to make. No timber, which is such a perishable material, is used, there being nothing in the shaft at all but three pipes, viz.: the pressure pipe from the accumulator house, the exhaust water pipe going back to the accumulator house, and the rising main or pump column; in fact, the pumping compartment, if made as large as is usual for the spear rod system, may be used for a hoisting shaft. There are no heavy moving masses in the shaft. In the six shafts named to illustrate this subject, the moving mass will not be short of 4000 tons, while in this system all the moving machinery, outside the accumulator house and excluding the bucket lifts, will not amount to 100 tons. This point cannot be too strongly urged. We do not believe that, with the weights in motion, necessary for depths over 2000 feet, that more than 6 strokes per minute can be made with the 14-inch pumps and 8 foot stroke. We have already had sufficient instruction on this point from a rather serious chapter of accidents. The inertia of the great weight of pit work at the beginning of the stroke and its momentum when in motion, being as we have already seen, fatal to the geared pumping engine, the hydraulic engine will make as many strokes safely at the fifth, or 3000 feet station, as it can at the first, or 600 feet station. The cutting and timbering required in the shaft, is far less for the hydraulic than for any other system now in use. The stations being 600 feet apart instead of about 250, as with the spear rod system, no large stations for balance bobs are required. The whole

underground work, so complicated in the present method of working, is simplicity itself; while the risks attending its working from breakages, are reduced to the strength of the pipes, compared to the innumerable connections and guides in the pump rods, and the sustaining and keeping in line heavy pump columns and pumps, at short stations, with the frequent examination of the same while at work, and besides all this there is the balance bobs in the vertical shaft and the V bobs in the incline, with all their attendant connections and bearings, to be kept in line, and heavy timbering required to secure the walls and roofs of so large stations.

As we have already pointed out the convenience of driving each pump at just the speed suitable to the quantity of water required to be raised from that particular station, or the convenience of attaching intermediate auxillary pumps at the point where water is found, is a feature of this system that should recommend it to favorable consideration. In the present system, the pumps are usually all one size, from the surface to the bottom; make the same number and have the same length of stroke. This necessitates in many cases a stripping of water back to the cistern, in order to keep the pumps in water, wasting, by this means, a great amount of power. To illustrate: suppose a shaft, with six sets of plunger pumps, total depth 1500 feet, there is a flow of water equal to 100 gallons per minute at the 500-foot level, 75 gallons at the 1000-foot level, and 225 gallons at the sump in the bottom. In this case, the two upper pumps have to deliver 400 gallons per minute, and must be run at the speed necessary to do that; the next two sets of pumps receive only 225 gallons from below, and 75 local drainage; but making the number of strokes required for 400 gallons, 100 gallons must be let back from the pump column to the cistern; the lower two sets of pumps, having only 225 gallons to deliver, must strip back 175 to the cistern. Here the effective duty equals 3838500-foot lbs., or 116-horses power; whilst the actual work done equals 4980000-foot lbs., or 151-horses power. The question might be asked, why not proportion the diameter of the plungers to the quantity of water to be pumped from the various levels. But this cannot safely be done where continual sinking is the rule; the water may at any time, while sinking is going on, disappear from the higher levels and be encountered at the bottom.

Now, let us suppose in this case that hydraulic pumping engines were used, one working on the 500-foot level, one on the 1000-foot level, and one on the 1500-foot level, all the pumps being of equal capacity; the upper pump having to deliver 400 gallons would make say 16 single strokes per minute, the next one having to deliver 300 gallons would make 12 strokes

per minute, the bottom one having to deliver 225 gallons would make 9 strokes per minute, thus the effective duty and actual work would be the same and 35-horses power saved.

The spear rod system is confined to the main shaft. Dip workings, incline drifts and prospecting winzes, must either be connected to the main shaft by a drain level, or, as is usually the case, pumped out by means of ordinary steam pumps worked by compressed air. This, as we have explained already, is a most extravagant way of pumping, yet it is about the only method of draining the workings under the present system. With the hydraulic system, the draining of any portion of the mine, however distant from the main shaft, forms part of the general plan. A small pipe which, in most cases, need not exceed one inch in diameter, is led from the main pressure pipe to the point to be drained, and there connected to a small hydraulic pumping engine; the size, of course, to suit the work to be done. This engine would pump through the drift to the next station above on the main shaft. The hoisting in winzes and prospecting inclines may also be done with the same power from the same pipes.

Should a large reservoir of water be struck, as occurred lately in the Savage mine, and it being desirable to pump it out as quickly as possible, a large temporary hydraulic engine can be placed at the desired level, and the whole reserve power of the accumulating engines directed to that particular locality and the reservoir drained in a very short time, while nothing of this kind can be done by the present method.

Should the general average of water in the mines worked by this system prove to be greater than was anticipated, other pumping engines can be placed in the accumulator house until the power exceeds the work, and this can be done without in any way interfering with those already at work. Accumulator rams can be added in the same way, if required. By the present system any addition to the power requires the reconstruction of the whole works.

While on the other hand, if the power provided for a certain number of mines is more than is actually required, the system can be extended to embrace other workings.

We have endeavored to point out some of the most important features of the hydraulic pumping system, wherein it has the advantge over the spear rod system. We do not know of any mechanical difficulties in such a scheme; the only objection that can be urged against it, is that of vested interests. What shall be done with all the present pumping plant should such a scheme be adopted for a number of the leading mines? We might answer this by

asking another question. What shall be done with it, anyhow, when it ceases to be able to drain the mines, either from an increase of water or the natural increase of depth? The time for this question is not so far away in the future as is supposed by some. It is now time for some of the mines mentioned in this scheme to answer it. By looking at the table of sizes given for certain depths and quantities of the differential compound pumping engine, it will be seen that the engines but lately put to work on some of the shafts will not pump to the depth that it is understood they are intended to pump to. We do not suppose that the engineers of these mines are deceived themselves in regard to the capacity of those engines, but they have allowed the companies interested to believe a good deal of nonsense regarding the depth to which they can go with their pumping plant. Time will settle the question of what shall be done with the present plant in some cases very soon. The question will then be, what shall be the future plant? Then we ask for a careful study of the hydraulic system by all those interested in the progress of the mines. We know that our views are opposed to those of some of the leading engineers on the Comstock, and also to our engineering brethren in the business of designing and building such machinery. Yet we will hold to our plan, and urge it as the best that can be adopted so far as the work to be done is concerned. Whether the interests of several companies can be united in any general drainage scheme is not the part of the problem that belongs to us to discuss. From a mechanical and economical point of view we know that the hydraulic system is far ahead of any with which we are acquainted, and we have studied all the plans in operation both here and elsewhere. We have given but a hasty sketch of our ideas, yet we think enough to enable mining engineers and superintendents to form their own opinions as to the merits of this or that style of pumping engine and pit work.

APPENDIX TO PART I.

INSTRUCTIONS TO ERECTORS.

[By the Patentees of the Differential Pumping Engine.]

Setting out Position of Engine.

It is the duty of an erector to observe special care in setting out the center line for his engine; and he is a slovenly workman and an unqualified erector who, when he finds that the work is a little out of its place, attempts to "spring it to," instead of bringing it to a natural bearing. If the bed is a little on the twist or out of level, then it must be lifted and properly bedded before finally being bolted down. Before screwing the holding-down bolts tight, and after the bed has been properly leveled and lined off, and the cylinders are found to take their proper bearing on the bed without any straining to bring the meeting surfaces together, then liquid cement should be well grouted under the bed where there is space to admit it, but the stone or concrete should be dressed as near as possible to receive the bed in the first instance. It is defective work to avoid the dressing of the stone by the use of grouting.

Clearing Ports.

Before the cylinder and valve-chest covers are put on, the steam and exhaust ports must be washed out. Cylinder covers have often been knocked out on starting engines, because a bolt or nut has been accidentally dropped into the steam port.

Washing out Pipes, etc.

All pipes must be washed out, that no dirt or sand may be allowed to get into the engine. Great damage is often done to an engine for want of this precaution.

Making Joints.

Avoid the use of red lead cement, except for permanent joints. Use india-rubber canvas for joints that are not scraped and require to be broken occasionally. Make the joints of steam chest, subsidiary cylinder, cataract, air pump, and stop-valve covers with india-rubber canvas.

Setting Pistons.

Set the pistons of the compound engines that they may be made to touch the ends of the cylinders at the same time. In working, the pistons should work $1\frac{1}{2}$ inches clear of the covers in small engines, and 2 inches clear in large ones.

Setting Valves.

Set the valves by putting the main piston and the subsidiary piston both in *exact* mid position. The mid position should be got by putting the piston first to touch one cover and then to touch the other, and dividing the length thus obtained. When the pistons are *exactly* in mid position, and the main valve lever is standing straight across the engine, put the main and subsidiary valves in *exact* mid position, and the valves will be properly set.

Packing Stuffing Boxes.

Pack the stuffing boxes of the cataract and subsidiary cylinder with good clean *spun yarn* and tallow: *never use any hard packing for these boxes.* Pack the boxes of the drain or relief valves with white cotton and tallow, and see that the spindles move freely after packing.

Cataract.

The cataract ports must be *well washed* out before the cataract is filled. About a quarter pint of *best olive oil* should be put in if it is a large cataract, and less if a small one, and then filled up with clean water. The cataract feeder should be filled with best olive oil or clean water.

Condenser.

If the pump has a condenser in the suction pipe, a vacuum gauge should be put on, and the hand-wheel screwed down till the gauge shows a vacuum of 9 to 10 lbs.

If the air pump is driven by the engine, and the condenser has to draw its water from the sump, a jet should be put into the condenser from the main column, or from any convenient supply where some pressure of water can be had to enable the vacuum to be readily formed in starting the engine, otherwise there will be sometimes a difficulty in getting the condenser to draw its water.

Condenser Valves.

When the valve seatings are held down by a center bolt, see that the thread of the bolt is cut far enough up; if it is not, the bolt may be screwed in tight without firmly securing the seating.

PUMPS.

Lift of Valves.

Give *single mitre valves* a "free" lift equal to *one-eighth the diameter of the valve.* If the valves beat heavily reduce the lift. Let *double beat valves* have a "free" lift equal to from *one-tenth to one-twelfth the diameter of the valve.* The lift of the valve should be less when the head of water is very great.

Air Cock.

Always put an air cock on the highest part of the barrel of the pump, bearing in mind that air always lodges in the highest available space.

Valves and Spindles.

Let the valve spindles fit very easily in the guides, and in screwing down the valve box covers make the joint come together, "face and face," otherwise the spindle guide

may be a little out of the vertical line, and if so, will cause the spindle to "bind" and keep the valve up. If the joint material does not allow the joint to come together "face to face," it must be pared or thinner material put in.

Suction Rose.

For all forcing pumps, the "wind-bore," or end of the suction pipe, should be surrounded with a wire basket, with not more than one-eighth or three-sixteenths of an inch mesh. If this precaution is not taken, the valves may be broken or the spindles bent by "gags" getting into the pump.

Stuffing Boxes.

Pack the stuffing boxes of the pumps with a carefully made and plaited gasket, steeped in melted tallow.

HYDRAULIC ENGINES.

Let the instructions given above for pumps be attended to, and *always couple the exhaust pipe from the engine to the delivery pipe from the pump.*

Drain Pipes and Valves.

Let the plugs be out of the drain pipes from the cylinders on first turning steam on to the engine, that the cylinders and pipes may be cleared of dirt by blowing through. The plugs are provided for the convenience of clearing the pipes in the case of choking. Sometimes they will choke with tallow. Let all the valves be connected to one pipe leading to the condenser. If not connected to the condenser, the low-pressure cylinder will not be drained.

Adjusting and Working the Engine.

There are two distinct motions combined to work the slide valve. One, the motion taken from the engine, and the other given by the cataract. To make the engine work uniformly, especially when unequally loaded, as often happens with sinking engines, two adjustments must be made, one with the stop valve, and the other with the cataract. *Let the stop valve and cataract be both closed before starting, then open the stop valve a little, and if the engine does not make a stroke sufficient to reverse itself, then open the cataract and stop valve gradually till it does do so.* It is important that the cataract and stop valve should *always* be adjusted to suit each other. If the engine does not reverse itself, it is because the stop valve admits too much steam and causes it to tend to race, and in doing so, it corrects the racing by over-running the cataract motion and reversing the steam to catch the piston. This often happens in sinking, from having a "riding column" on one of the lifts, or because the lifts are not balanced. *It is also important to see that the cataract does not work too fast* for the speed of the engine; if it does, it will not have proper control of the engine. If there is a knock at the end of the stroke of the subsidiary piston, it is an indication that the cataract is working too fast, and it should be closed a little.

Before starting the engine for the first time, see that the *crank pin* on the *rocking shaft*, which drives the main valve lever, is in its *highest position, and the cataract and feeder filled* and the cataract closed. Start the engine, and if, *after the load is on, and*

the proper vacuum has been obtained in the condenser, the engine does not make its proper length of stroke, then lower the crank pin till the proper stroke is obtained. *Always* observe that the *load and the full vacuum are on the engine before adjusting the stroke*; because, if this is not done, the engine will lengthen its stroke when the vacuum and load are on. Should the pistons work nearer one end of the cylinder than the other *when the load and vacuum are on*, the adjustment must be made on the screw of the connecting-rod between the rocking shaft and the valve lever. *If the pistons work nearer the front cover, then lengthen the rod*, and if nearer the *back cover* then *shorten the rod* a half turn or a turn of the screw as may be required. If this does not answer, then it is a proof that the valves are not properly set, and the instructions above given for setting them must be attended to.

DIFFERENTIAL STEAM PUMPS AND SEPARATE CONDENSERS.

Pack the glands of the subsidiary cylinder, cataract, and valve spindle with clean soft spun yarn and tallow.

Fill the cataract and feeder with best olive oil and water.

To lengthen the stroke of the piston screw *down* the *eye* that drives the main valve, and to *shorten* the stroke screw it *up*. If the piston works too near one end of the cylinder, it can be adjusted by means of the nuts on the *subsidiary and cataract rods*. If the piston works too near the *front* cover, screw the *cross heads back* towards the cylinder; and if too near the *back* cover, then screw them *forward* towards the pumps.

If the steam pump has a tappet rod, the tappets must be fixed in such a position that at each end of the stroke the hand lever has a little play. If this is not attended to before starting the engine the rod may be broken.

RULES FOR THE ENGINE ROOM.

1. A place for everything and everything in its place.

2. A use for everything and everything to its *proper use*. *Never use a spanner for a hammer or a hammer for a spanner.*

3. Keep the spanners neatly arranged on nails on a board fixed on the face of the engine room wall; arrange the spanners on the board in order or to a device; it will look well, without involving any extra trouble, and will show at a glance if one is missing.

4. Be provided with a few hard wood packing sticks for packing stuffing boxes with, a lead hammer, a steel hand hammer, a short bar of copper for driving out pins with, one or two hand chisels, an adjustable spanner, a center punch, an oil can, a tallow kettle, and some bits of wire to clean out oil holes with. Keep these things in a locker and *never lend them to anybody*.

5. Take a pride in keeping your engine clean and in thorough order (without a hammer mark), by attending to every want as soon as discovered. Your work will be thus greatly lightened, and you will get the CREDIT OF BEING UP TO YOUR DUTIES.

6. Let the *stop valve* and *cataract* be *both* closed before starting, then open the *stop valve* a little, and if, after the engine is warmed, it does not make a stroke long enough to reverse itself, then *open the cataract gradually* till it does do so. Be careful that the cataract is not made to work too fast for the speed of the engine. If it is, it will not control the engine. When working too fast, there will be a knock in the subsidiary cylinder.

7. See that the cataract does not work too fast; if it does, it will not have control of the engine, and if it works too slow, or if the stop valve is open too wide, it will check the engine too much.

8. If the engine makes too long a stroke, raise the crank pin in the lever of the rocking shaft which drives the main valve lever. If it makes too short a stroke, lower it; but before making any alteration, see if the correction is required when the full load and full vacuum are on the engine: *if not, it must not be altered.*

9. If the piston works *too near* the *back end* of the cylinder, *shorten* the connecting rod of the valve gear by the screw, and if too near the *front end* then *lengthen* it, a half-turn or a turn as may be required.

10. *Never alter the engine for the sake of experiment.*

11. It often happens in sinking that by getting the "lifts" greatly out of balance the pistons will be caused to work nearer one cover than the other, and on that account it is advisable to work the engine at a shorter stroke in sinking; but by keeping the "lifts" nearly balanced, *as they should be*, the engine is kept in its proper way of working.

Part Second.

HOISTING MACHINERY.

During the earlier period of the mining history of this country, the gold was gathered, as is well known, from the surface deposits of our placer diggings. The precious dust then collected, was gathered from river bars and beds, dry ravines and alluvial flats, or similar locations, constituting what is called washings or stream works. These gold bearing deposits of alluvial and diluvial drift, are mostly formations caused by currents of water and such other physical forces as are active at the present day. Being on the surface, the working of these deposits, are matters so simple, and of such easy accomplishment, as to preclude them from being classified under the head of regular mining operations, though while virgin, they have often yielded munificently upon a slight application of labor and capital; being shallow, however, and generally of narrow limit, they are subject to rapid depletion; this class of diggings in California having become, in a few years after their discovery, so far impoverished as at one time to awaken fears that gold mining would have ceased to be carried on as a profitable business in the State. Ere these surface accumulations became exhausted, they had yielded the means whereby research was made after new forms of deposit, and measures taken for penetrating and laying open the old to a greater depth. Fortunately, these efforts, though sometimes resulting in failure, have, in many instances, been attended with success. These deep lying beds of cement, gravel, and auriferous earths, have necessitated shafts and hoisting works; and the still deeper quartz veins on the Comstock, have called for considerable engineering to meet the demand for powerful hoisting works, although the difficulties are not so formidable in deep hoisting as in deep pumping works, yet still they are sufficiently so as to require the careful study of the mining engineer.

First-class shafts are now generally sunk to a depth of one thousand or twelve hundred feet, by what might be termed temporary machinery; this differs little from the permanent works for shafts, whose ultimate depth is not to exceed one thousand feet, unless it be in position; sometimes if the

are very much alike on all the mines, the only difference being in size. The plan of pumping and hoisting, shown on Plate 3, represents the most recent constructed hoisting works of this class. In this particular case, the engine, as will be seen, is horizontal; cylinder 20 inches diameter and 36-inch stroke; the fly-wheel is turned up and fitted with a brake; there are two pinions fitted to the engine shaft, 43½ inches in diameter, 3½-inch pitch and 9-inch face; the teeth are capped or flanged to the pitch line. These pinions run loose on the shaft, and are fitted with sliding clutches, operated from a platform not shown in our engraving. There are two separate reel shafts, each carrying a spur wheel, reel and brake wheel. The spur wheel is 120 inches in diameter, 3½-inch pitch and 9-inch face, flanged to the pitch line. The reel center is 4 feet in diameter, with arms as long as will run clear of the pinion clutch, and wide enough between to take a 4½-inch by ⅜-inch flat rope. The brake wheel is 10 feet in diameter and fitted for a brake band 8 inches wide. The pit head shieves for this class of hoisting machinery are generally made about 8 feet in diameter. Few of the hoisting works differ materially from that we have just described. Few of the mining superintendents favor the plan of working two ropes balanced, the objection being the difficulty of working from different stations in each shaft, although we think this objection might be partly overcome by running past or overlapping the stations. If the objections to working the cages balanced can be satisfactorily overcome, then the clutches, which are always more or less a source of trouble, can be dispensed with and direct acting hoisting works, instead of being the exception, would become the rule. When the hoisting works of the Consolidated Virginia and California combination shaft are permanently at work, extracting ore from a fixed point, it will not then be necessary to use the clutches except to adjust the length of rope. The advantages of the direct acting system will then be seen.

We have already stated, that in vertical shafts, flat ropes are all but universally used, and while the system of working each reel independent is adhered to, flat ropes have the advantage. The machinery is more simple and compact, which means that it is also cheaper than for round rope, that is for great depths. The rope winding on itself, gives the reel an ever varying diameter, the engine thereby gaining leverage as the load increases. To work a shaft of, say 2500 feet, by round rope, requires winding drums of large diameter to get the width within working limits; this makes gearing necessary, and if the ropes have to run independent of each other the great weight of the winding drum makes the starting of it, through a clutch, a rather serious matter; yet the short time that a flat rope lasts and the con-

stant repairs it requires, while it does last, has led to a desire among mining superintendents for hoisting work using round rope, having besides all the advantages possessed by those using flat ropes. We need hardly state that this has not yet been accomplished, but it no doubt will be when demanded. In most of the geared hoisting works, the engine cylinder is used as a species of air brake; in lowering, the reversing link is notched up or set, for running the other way, air is drawn into the cylinder from the exhaust and compressed between the piston and cylinder heads; the pressure brought on the cylinder and passages in this way, must be very great. The brake is effective, but we cannot help thinking that the system is, to say the least, rude and destructive to the machinery. The brake power required when running, unbalanced and a great weight of rope added to the weight of cage and car is very great, and any plan that would enable hoisting works to be run balanced and at the same time keep the ropes independent in their motions, and thereby save the great waste of power, which means fuel, attendant on the present system, will help the economical development of the mines. We are engaged now in perfecting such a plan, but have not yet advanced far enough to publish anything on the subject, though we trust ere long to be able to do so. The excessive high speed of most of the geared hoisting engines on the Comstock, is very destructive, both to engines and gears. We know one instance of a 20″ by 36″ hoisting engine making regularly 175 revolutions, which is equal to a piston speed of 1050 feet per minute. No engine or foundation can long stand the strains due to such a speed as this.

The first of the incline hoisting work, such as that at present working the Belcher incline, are similar to those for working the vertical shaft; but when it was determined to sink these mines to the 3000-foot level, the consequent length of incline made larger incline hoisting works a necessity. The incline hoisting works of the Crown Point mine was the first of the larger class. I. F. Thompson, Esq., who was then consulting engineer to the Crown Point Mining Company, determined to use a large winding drum and round rope, the drum to wind 2100 feet of rope. These works were built by the Risdon Iron Works, in the year 1873, and are of the following style and dimensions: Two horizontal steam engines, set 22 feet apart, with their shafts coupled together and cranks set at right angles; diameter of cylinders, 20 inches; length of stroke, 42 inches. These engines are handled by the ordinary link and reversing lever. The fly-wheels, of which there are two, one on each side of the reel bed plates, are 16 feet in diameter, and weigh 18000 lbs each. These wheels are turned up and fitted with a brake. The bed plates that carry the drum shaft extend back under the

OPHIR G.& S.M.CO.

GENERAL PLAN & ELEVATIONS OF

ENGINES, HOISTING REEL & WATER BALANCES.

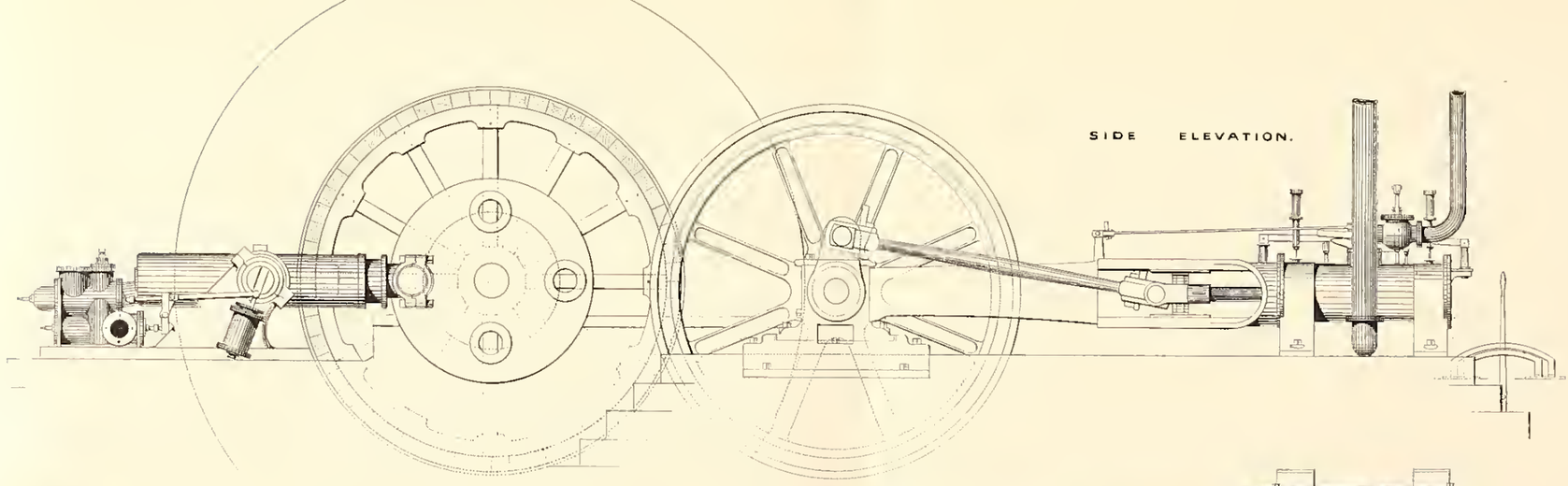
BUILT BY THE

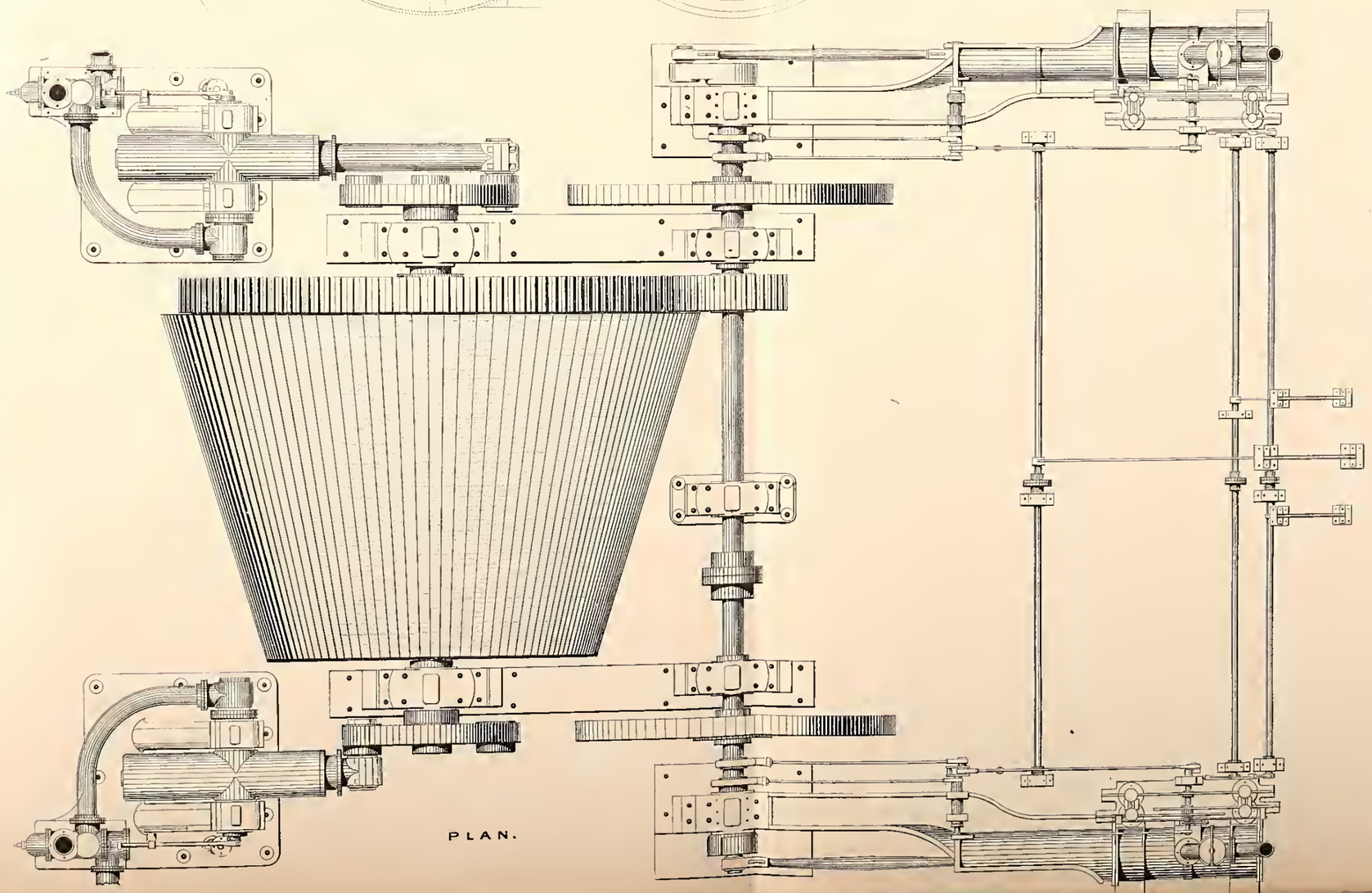
RISDON IRON WORKS, S.F.

JULY, 1875.

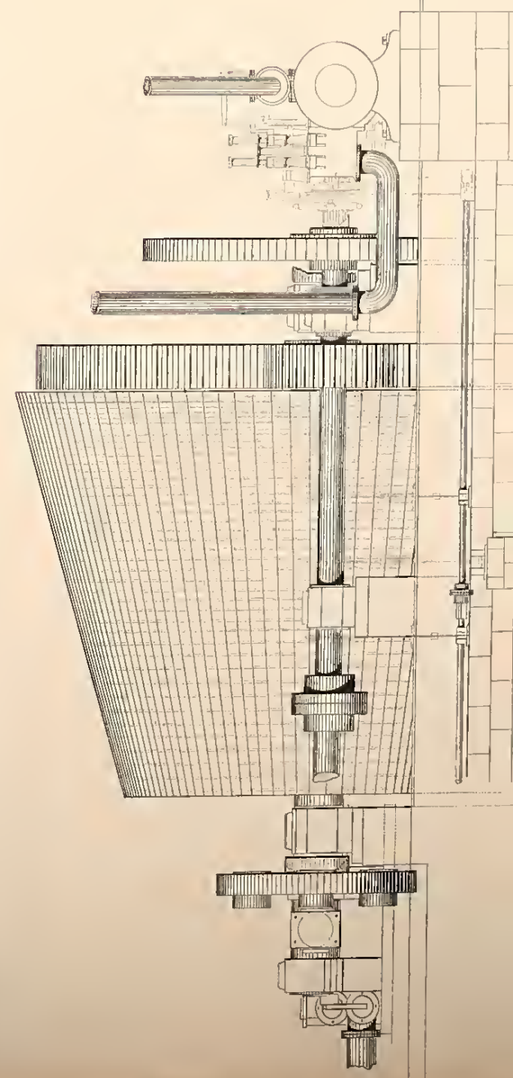
SCALE.

SIDE ELEVATION.





PLAN.



engine shaft and carry the intermediate bearings for it. The spur pinion, keyed to the engine shaft, is 55⅜ inches in diameter, 6-inch pitch and 19-inch face. The engine shaft is 9½ inches in diameter, and the drum shaft is 13 inches in diameter. The spur wheel, to which the drum is secured, is built in segments, and is 252.1 inches in diameter to the pitch line, 6-inch pitch and 19-inch face. On one side of the rim of this spur wheel a heavy flange is cast to take the one side of the drum; the rest of the drum is supported by cast iron rings, with wrought iron arms, bolted to centres on the shaft; the outside of the drum is formed with hard wood staves, 6 inches in thickness; this, again, is covered with cast iron plates with grooves to fit the rope; the drum is 6 feet, 1½ inches in width, and 21 feet in diameter: the rope is a steel cable, 2 inches in diameter, and the drum takes 33 turns.

These hoisting works have done very well so far as working goes, although a slight increase of engine power would be an advantage in starting the load, as to economy in working this design leaves much to be accomplished. The hoisting works being on the surface there is a dead load of rope, equal to 7000 lbs, always hanging in the vertical shaft, besides the weight of cars and the rope in the incline unbalanced, making the actual work done a very small part of the power used. In other words, the greater part of the power developed in hoisting is absorbed in brake friction in lowering.

The other two large hoisting works of this class on the Comstock, that of the Savage and the Ophir mining companies, of which I. F. Thompson, Esq., was also engineer, were built in the beginning of 1875. The Savage works were built by the Union Iron Works, and the Ophir by the Risdon Iron Works. These two hoisting works are in their main features alike, the slight difference in the style of engines is only the result of a difference in taste in the designing departments of the two shops. The Ophir works, however, possess one distinct feature, that of a hydraulic balance for the weight of rope in the vertical shaft and the weight of car in the incline; outside of this balancing arrangement, the description we give of these works, will apply either to the Savage or the Ophir. The winding drums are placed about 120 feet back from the shaft; this distance back is necessary, owing to the width of drum, requiring the rope to deflect 7 feet on each side of the centre line. Plate 25 is an elevation and plan of the Ophir works, which is clear enough to be understood without any description. The engine-power consists of two horizontal non-condensing engines, diameter of cylinders, 24 inches, length of stroke, 48 inches, with link reserving gear. The valves are double beat mitre valves, with Cross's variable cut-off. The engines rest on feet cast to the cylinder and the main

pillow block, these two points being connected by a frame, which forms the guide for the cross-head. This style of frame having been adopted by Mr. Corliss for his horizontal engines, it is sometimes called the Corliss frame. The engines are placed about 30 feet apart, the shafts being coupled near the centre, the cranks are set at right angles, there are two fly wheels, one near the main pillow block of each engine, the main bed plates that carry the bearings for the winding drum extend under the engine shaft and carry bearings for it; there is also a centre bearing for the engine shaft, making in all five bearings to the shaft. The spur pinion is the same as that of the Crown Point works, being $55\frac{3}{8}$ inches in diameter, 6-inch pitch, and 19 inch face; the engine shaft to which this pinion is keyed, is 11 inches in diameter. The spur wheel is also from the same pattern as that of the Crown Point, being 252.1 inches diameter to the pitch line, 6-inch pitch, and 19-inch face. The drum shaft is 14 inches in diameter, and 17 feet long between the bearings. The drum is built tapering, having a diameter of 23 feet at the large end, and 14 feet 3 inches at the small end, and is 14 feet 6 inches in width, and will wind 4000 feet of $2\frac{1}{4}$ inch diameter round rope. The large end of the drum is carried on a cast iron ring, which is bolted to flanges cast on the segments of the spur wheel; the rest of the drum is carried on three cast iron rings, one at the small end and two intermediate; these rings are supported by wrought iron arms, bolted to cast iron flanged centres, keyed on the shaft. The outside of the drum is built of white mahogany staves, 6 inches in thickness, outside of which is a covering of cast-iron plates, made with grooves for the rope; the pit head shieves are 14 feet in diameter and are built in segments, with wrought iron arms.

The hydraulic balance, which, as we have already stated, is applied only to the Ophir works, there being 1500 feet of rope in the vertical shaft at this mine, the need for some such appliance was more apparent. This water balance, as will be seen from our engraving, consists of two hydraulic engines applied directly to the drum shaft. During the operation of lowering, instead of taking, the descending weight of rope, etc., on the brakes, it is taken by the hydraulic engines, which for the time are acting the part of pumps and forcing water from a tank in the engine house, through a line of ten-inch pipe up the mountain side, at the back of the works to a reservoir, constructed at an elevation of 400 feet above the hoisting works. When the motion of the drum is reversed for hoisting, the pumps become hydraulic engines, and the power that was stored up in lowering is used to help the steam engines in hoisting, or it may be used for other purposes, as we will

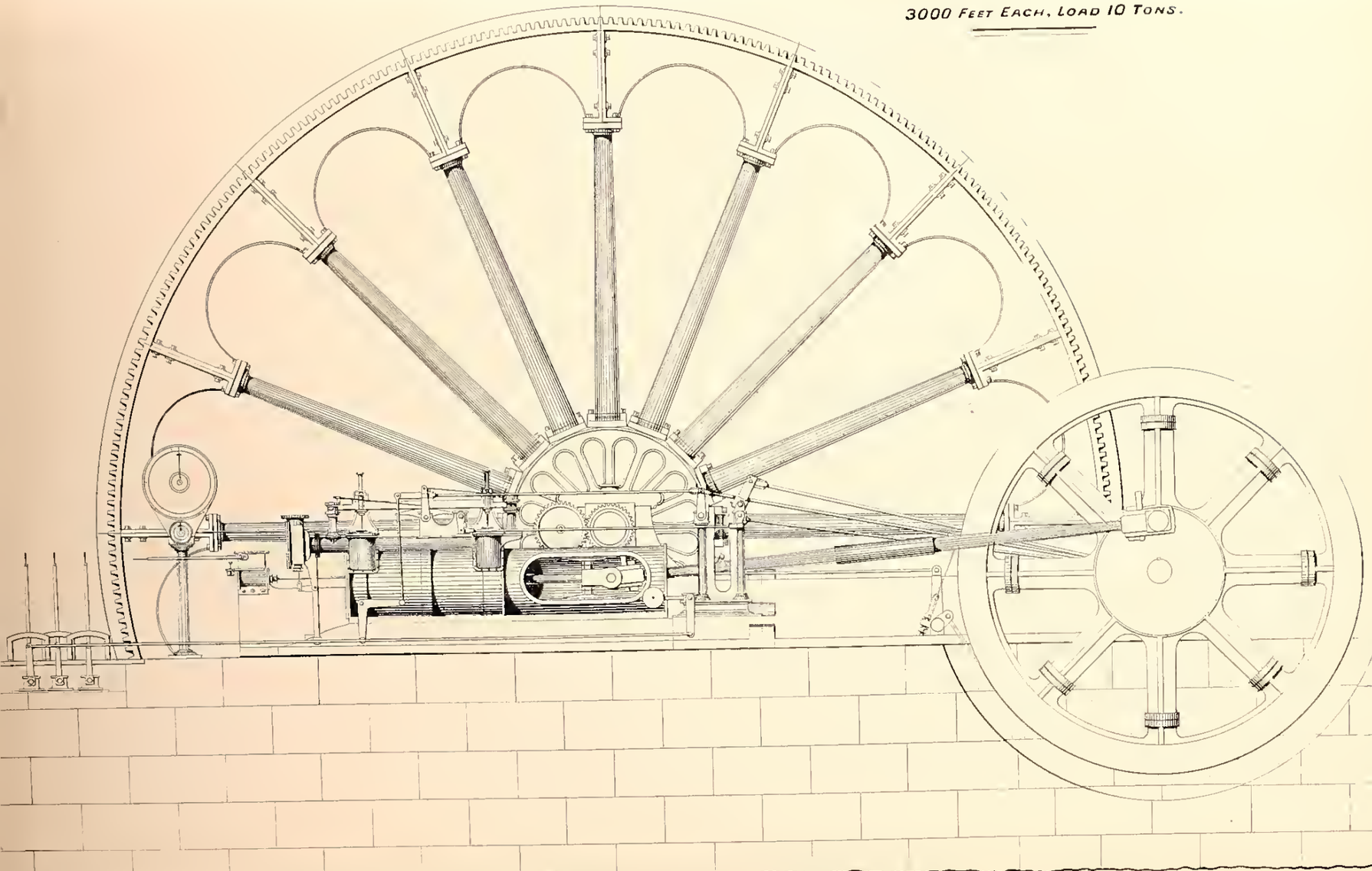
INCLINE

HOISTING WORKS.

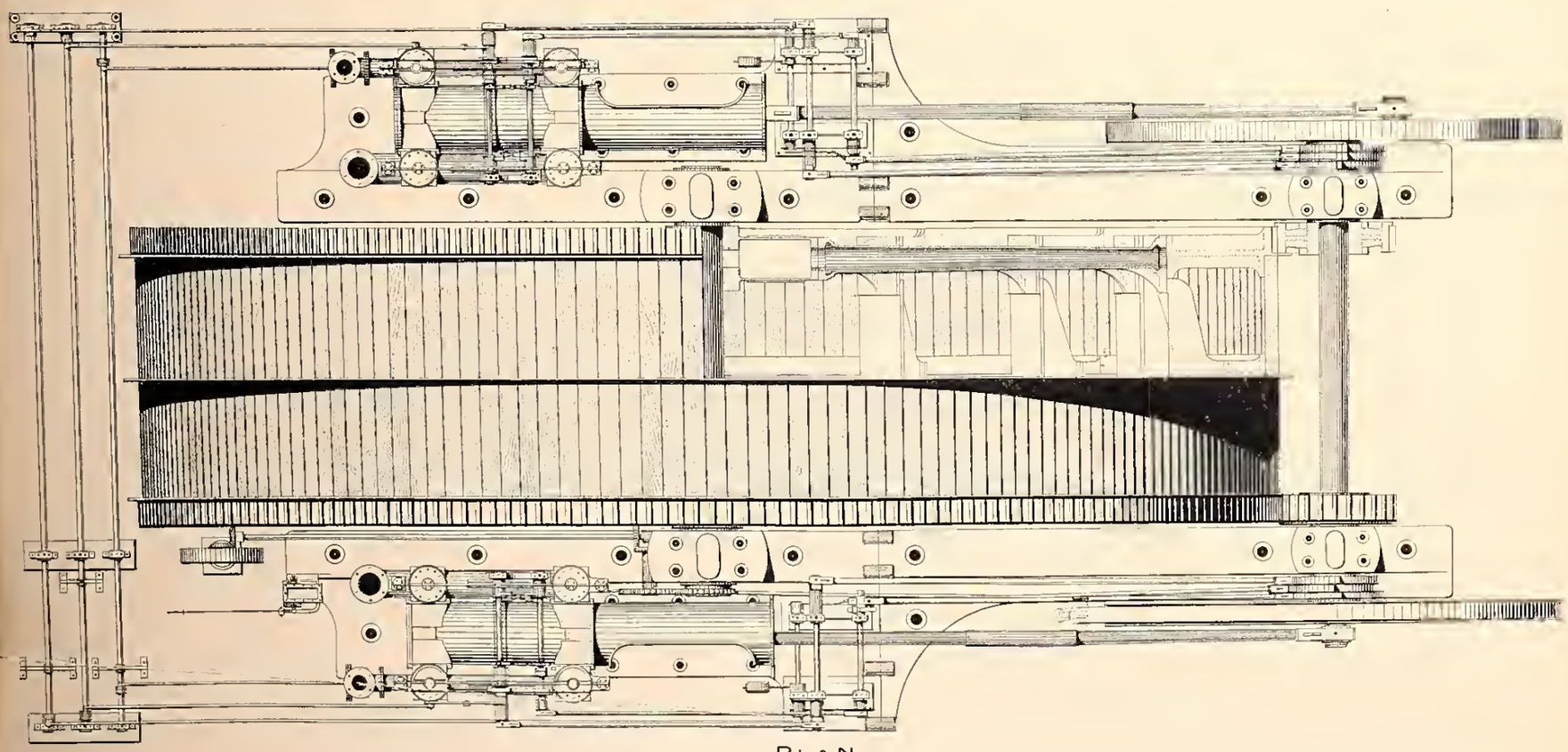
FOR

TWO BALANCED ROUND ROPES

3000 FEET EACH, LOAD 10 TONS.



ELEVATION.



PLAN.

SCALE

see further on. On each end of the drum shaft a large cast iron crank disc is securely keyed. These crank plates are 91 inches in diameter and have each four crank pin holes, bored so that the crank pin will fit either of them; this admits of the stroke of the rams being either 54 inches, 60 inches, 66 inches, or 72 inches, according to the amount of dead load requiring to be balanced. The crank pins are 9 inches in diameter and 16 inches length of bearing; the ram heads take hold of the pins directly. The rams are 18 inches in diameter and of sufficient length for a stroke of six feet. The cylinders are carried by trunnions set in the center of their length. These trunnions are 14 inches in diameter and have a bearing of 16 inches. The water is introduced and discharged through one of the trunnions. The valves are operated from a pin on a sliding plate that is fitted to the other trunnion; this sliding plate is attached to the ram of a small hydraulic cylinder, by which the engineer can, from his platform, at once slide the pin to the center of the trunnion, and the hydraulic balance is thrown off. The bed plates, which are massive castings, have the trunnion blocks cast upon them. The valve arrangement differs little from that used by Sir William Armstrong in his largest hydraulic engines. The whole work is set on massive foundations, and is, without doubt, the finest incline hoisting works in this country. We have stated that the power stored up in lowering might be used for other purposes. The ore extracted from the Ophir mine is sent by rail to the mills; and as the railroad reaches the Ophir at an elevation of about 40 feet above the works, the ore requires to be lifted about 56 feet to the bin, from whence it is dumped into the cars. A hydraulic elevator was designed by us and built by the Risdon Iron Works for that purpose; it has two balanced rams working two balanced cages, which elevate the cars from the level of the shaft to the ore bin above. The working of this machine is perfect, and uses but a small portion of the water pumped up by the balance rams on the incline hoisting works. It will be observed, however, that the position of the Ophir works is very favorable to the working of a hydraulic balance, and there are localities where this plan would not be suitable; besides, the works are very expensive. We are inclined to think that for inclines it would pay to sacrifice some of the convenience of working independent ropes, in order to reach the greater economy of working two ropes balanced. We have designed and illustrated, on Plate 28, an incline hoisting works, working two balanced ropes and winding 3000 feet of rope; this arrangement is so clearly illustrated by our elevation and plan that we need hardly describe it. As will be seen from our engraving, there are two massive bed plates, which

serve as foundation plates, both for engines and drum. The engines have been designed to meet the special requirements of their position, and have cylinders of 24 inches diameter and a stroke of 48 inches, and are fitted with a variable-cut off. The fly-wheels, which are hung outside the bearings, and serve as crank plates, are 16 feet in diameter and will weigh 20,000 lbs. each; one of them is used as a brake wheel and is fitted with a steam brake; the eccentrics are all carried on the hub of the fly-wheel. The engine shaft, which is 11 inches in diameter, carries two spur pinions, which are keyed on close to the main bearings; these pinions are 48 inches in diameter at the pitch line, 4-inch pitch and 14-inch face, and gear with the spur wheels that form the sides of the drum. The spur wheels are 40 feet in diameter, 4-inch pitch, 14-inch face, and are built in 16 segments each; the hubs for these wheels are large hollow castings, to give room for the bases of the 16 columns that form the arms; these columns have brackets on their insides, facing each other, to which the beams are bolted that carry the center arch or ring; the segments of the spur wheels have flanges cast on their insides, to which the wood staves are bolted; no iron covering would be required for this drum. The diameter of the drum would be 40 feet and the width of each side 53 inches. We are satisfied that this arrangement would prove economical in working and would not cost any more for two ropes than the others have cost, which can only use one rope, while owing to the arrangement of engines the foundations would be less costly. As to the relative merits of the various forms of hoisting machinery, there are circumstances in every case that require special treatment, and what might be best for one mine would not be the proper thing for another. Having, therefore, given a few examples of successful works, we would leave it to the judgment of the mining engineer or superintendent to decide what is most suitable to meet the requirements of the particular case they are called upon to provide for.

At all the principal shafts, hoisting works are required for the pumping compartment. These hoists or winches are used for moving the pumps, spear rods, and other heavy weights that require to be handled in the shaft. In most of the geared hoisting works, these consist simply of a reel for hemp rope placed behind the main spur wheels of the pumping works, and having a wheel gearing into the pump wheels, the pillow blocks being arranged to slide the reel out of gear when not required; this arrangement is shown on Plate 3. The great objection to this plan is, that it can only be used when the pump gears are in motion, whereas it is generally needed when the pumps are stopped for repairs, or the moving of the sinking

column; this inconvenience makes this arrangement of little value where weights have to be frequently handled, and has led to the adoption of independent hoists for this purpose at all of the new works. On Plate 21, we illustrate a compact hoisting machine for round hemp rope to lift ten tons, designed by us; it being arranged to use the reels at present in use at so many of the mines. This machine would occupy little space and fill a want long felt; besides, it can be employed in lifting weights all over the works. On Plate 24, we illustrate the pumping compartment hoisting works at the Chollar, Norcross, and Savage combination shaft. This is the most powerful hoist of its kind in the country, and will lift, with the rope all wound on, 40,000 lbs., and with the rope all out, 60 tons. The rope is of steel, $7\frac{1}{2}$ inches wide and $\frac{3}{4}$ of an inch in thickness. The pit-head pulley is 20 feet in diameter, and from it the rope passes over the top of the cylinders of the pumping engine, behind which there is another pulley carried on cast iron columns, after passing over which, it descends to the hoist. Our illustration of this hoisting engine is so clear, that little description is needed to make it understood. The two side frames are cast in one piece, and carry the bearings for all the shafts and the engines. The two engines have cylinders 12 inches in diameter and have a stroke of 16 inches. The cranks are set at right angles; the crank discs being made large to serve as brake wheels. The engine shaft, as will be seen from our engraving, carries two pinions, one keyed fast to the shaft and the other fitted with a sliding clutch. The first pinion gears with the spur wheel on the intermediate shaft, which is 10 feet in diameter; the other pinion gears with the main spur wheel. The intermediate shaft also carries a spur pinion, fitted with a sliding clutch, which gears with the main spur wheel; on each end of the intermediate shaft a winch end is fitted. The hub of the main spur wheel is made 56 inches in diameter and forms the reel centre; the diameter of the spur wheel is 15 feet 8 inches, and the reel arms extend to the outer circumference of the wheel and winds 3000 feet of rope. This hoisting machine, as will be seen, can be run either in single or double gear; and in double gear, with the reel full of rope, will lift a load of 40,000 lbs., and with the rope all out, will lift a load of 60 tons. The working platform is placed 9 feet above the floor, and from it all the movements of the machine are regulated. This is an excellent arrangement of hoisting works, and will do all the work for which it has been built; but we think that for the ordinary work required for a hoist of this description, that a much smaller machine might be used, such as could be built at a price within the means of most of the mining companies, by using a four-fold tackle in the shaft for

lifting the heavier weights, such as the sinking pump with its column, and hooking the end of the rope on to the fauld of the tackle; or where hydraulic pumping machinery is adopted, the lifting of heavy weights in the shaft would be accomplished by simple hydraulic hoists. In fact, we have been for some time drifting into the conviction that if hydraulic pumping machinery should be adopted at any of the mines and an accumulator house established, from whence power can be readily distributed, that this method of transmitting power will not end in merely pumping out the water; but that hoisting, in inclines and winzes, as also in the main shafts, will all be accomplished by using water as a medium for transmitting the power; it is in this direction we think that the possibility of working independent ropes, any one of which can be operated without moving the other and yet balance all the load that is not delivered. When we consider that the weight of the rope, the cage and the car, is generally five or six times as much as the weight of rock raised, the saving that would be effected by any plan that would accumulate the power developed in lowering this great weight to be used again to assist in hoisting, would be worth the remodelling of all our hoisting machinery. In the deep quartz mines of Nevada, and especially those on the Comstock, there is a great field for the engineer. The fuel required to work those mines is simply enormous. Every year this fuel is getting scarcer; while the increase in the number of workings and the increased depth of those already in operation, adds to the quantity required. Every device, therefore, that gives promise of saving fuel, whether it be in pumping water, hoisting ore, or compressing air—in a word, whatever will lessen the power required for any of the operations carried on at the mines, either by saving of power, as in balancing hoisting works, or in economy of producing the power, as in expansive working of steam and air, will make the developing of these mines at greater depths not only possible but practicable. One of the objects we have had in view in writing this short treatise on pumping and hoisting machinery, has been to awaken an interest in these subjects amongst mining superintendents and those who have important interests centered in those mines. Economical working machinery in every department must become an absolute necessity, if great depths are to be reached. Never before in the history of those mines has this subject received so much notice by those interested as now. The stubborn nature of the water struck in the lower workings of some of the leading mines has created a desire to know what so much power costs in fuel and where it is possible to effect a saving. We have already pointed out that a careful record kept of the power developed

and the fuel burnt at the various mines, would help to establish something of a standard of efficiency, to be attained in all projected works and serve also as a check on wasteful systems of working. Whether it is possible, where so many conflicting interests are at work, to obtain a true and faithful duty record from each of the mines, is a question that can only be answered after trial; that it would be productive of good in drawing attention to the most economical class of machinery that is at present in use, we have not the slightest doubt, while it would also give to engineers the data from which future improvements could be deduced. We trust that this subject will be taken up by the superintendents at the various mines, and the proper data kept, from which to prepare tables of estimates per cord of wood for the various classes of machinery running; and where this is done, we would advise its being done thoroughly by competent men with the proper appliances, as data of a doubtful character would only mislead. In concluding this short and incomplete treatment of a very important subject, we would again state that our object has not been to thrust forward our own opinions and practice, but to claim for the whole subject embraced in our remarks a candid and careful study from those interested, either as superintending the working of mining machinery, or controlling the vast money interests represented by the mines.